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Proceedings of the Workshop on Acoustic Attenuation Materials Systems

National Materials Advisory Board

Commission on Sociotechnical Systems

NMAB-339

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current problems in the control of acoustic energy are those associated with such diverse applications as ultrasonic devices, space vehicles, and deep-diving oceanographic vehicles. Each of these, of course, may also have other quite different requirements of the acoustical materials in such properties as density, pressure response, and flammability.

The National Materials Advisory Board is frequently consulted on various specialized aspects of this broad field. In response to these inquiries, studies are organized and conducted by a small multidisciplinary group of experts. It is frequently useful for them to consult with other experts in fields related to those of the members of the group. Although this may be done on an individual basis, an open workshop with prepared papers and ample discussion time is desirable, whenever possible, to ensure that various viewpoints, experiences, and capabilities are represented. Thus this workshop on Acoustic Attenuation Materials Systems was organized and held on April 4, 1977, at the National Academy of Sciences.

The workshop program was assembled by a steering group, the roster of which is listed on page v. This group is an adjunct to the Committee on Structural Application of Syntactic Foam, whose classified study required that a workshop be conducted to explore the application of composites utilizing syntactic foams for acoustical absorption.

The program for the workshop was organized into two major sections, namely, formal presentations by invited speakers on the two topics, Materials Sciences and Applications Technology, followed by a discussion session. Samuel Dastin, the discussion rapporteur, conducted a lively session with sufficient time for audience comments.

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Proceedings of the Workshop

on

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PREFACE

The choice and use of materials for acoustical attenuation is an important technical problem as well as of great interest for improving comfort and working efficiency. Although the major effort in this field is at ambient conditions and in the audio frequency range, there are important applications that involve widely varying pressures, temperatures, and frequencies along with other requirements peculiar to the particular use. Among current problems in the control of acoustic energy are those associated with such diverse applications as ultrasonic devices, space vehicles, and deep-diving oceanographic vehicles. Each of these, of course, may also have other quite different requirements of the acoustical materials in such properties as density, pressure response, and flammability.

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The papers presented by the speakers and the comments during the discussion are included in the following pages.

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MATERIALS SCIENCES SESSION

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INTRODUCTORY REMARKS

Acoustic attenuation, i.e., the reduction of sound intensity transmitted from one place to another, has become an increasingly important issue in many fields and industries. Of particular importance is sound attenuation in industrial applications such as insulation for residential, office and industrial buildings, transportation (automotive, aircraft, and ships), and aerospace. The efficiency of sound attenuation depends greatly upon the audiofrequency range which varies greatly in these applications. Polymers, and in particular cellular plastics and elastomers (foams), either with or without fillers have successfully been employed for this purpose.

Syntactic foams which consist of dispersions of hollow glass or plastic microspheres of varying diameters in a rigid plastic matrix which may contain elastomeric components and other fillers (metallic or inorganic), have proven to be of special interest in this field as their various applications in industry as well as by the Navy testify. The usefulness of these foams is not only due to their ability to reduce the sound intensity but also to the fact that they are able to withstand severe environmental conditions and at the same time exhibit good mechanical strength.

The first part of this Workshop on Acoustic Attenuation Materials Systems is the Material Sciences Session and is devoted to a discussion of the fundamental scientific aspects of acoustical sound attenuation together with a review of the influence of materials, especially of polymers and their relationships of various physical to acoustical properties.

MECHANISMS OF ACOUSTICAL ABSORPTION

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1. INTRODUCTION

This paper is a survey of some of the more important mechanisms of absorption and attenuation of acoustic waves. We will review the acoustics of the processes involved, provide physical insight, and note both similarities and differences that one finds for waves in air and waves in water. We shall point out orders of magnitude of the quantities we are dealing with, giving rules of thumb where possible.

In a survey discussion such as this it is necessary to make some compromises both in scope and in depth of coverage. However, our intent is to provide a general introduction to what follows in the workshop, particularly for those who are new to the subject, or who have been away from it for some time. From time to time the reader will be asked to "shift gears," as we take up a new topic, or consider an alternative view. The treatment will touch on the following:

- absorption of acoustic waves at a surface,
- absorption in gradual or tapered transitions in acoustic properties,
- attenuation within a dissipative medium,
- structural damping - particularly, a review of several important mechanisms for the damping of flexural waves.

Waves and Wave Types

In an extended medium having both elasticity and inertia (i.e., capable of storing, at least momentarily, both potential and kinetic energies) there is the possibility of wave propagation. What is required is a proper connection (interaction) between the elasticity and

inertia, a constructive "cooperation".* Examples of wave-bearing systems are the following, with potential and kinetic energy storage mechanisms, respectively, indicated:

Transverse waves:

- waves on a string (tension; mass per unit length)
- surface waves on water (gravity; fluid density)
- internal waves: interface between oil & vinegar (gravity acting on differential density of fluids; fluid densities)
- flexural waves on plates & beams (flexural stiffness; mass per unit area)

Longitudinal waves:

- in fluids - i.e., acoustic waves (bulk modulus; density)
- in solids - again acoustic waves (appropriate elastic modulus; density)

Shear waves in solids: (shear stiffness; density)

The resulting wave, of whatever type, at a given frequency f travels at a certain speed and will have a wavelength λ such that

$$c = f\lambda. \quad (1)$$

As is indicated in Figure 1 the wavelength is simply the distance between points of equal phase. The wave amplitude quantity represents whatever measure of the motion is of interest, e.g., displacement, velocity, acceleration, pressure, stress, force, etc.

* Usually we think of continuous media in connection with wave propagation, but under the right circumstances waves can also be propagated in a ladder or network of discrete structural elements. Again the proper interrelation (interconnection) between elasticity and inertia permits wave propagation. Examples include macro-composite materials and structures, trusses, ribbed plates, etc.

The amplitude of a wave may vary with position either a) for geometric reasons, as in divergent or convergent waves, or b) because energy is removed from the wave, as by dissipation. Clearly both effects could be present simultaneously.

2. ABSORPTION OF PLANE ACOUSTIC WAVES AT PLANE SURFACES

Let us consider that the wave sketched in Figure 1 represents an acoustic pressure wave in a fluid. The wave advances with time, as is indicated by the dashed curve. For small-amplitude disturbances the acoustic pressure p and the fluid acoustic particle velocity u are linearly related, via

$$p = Z(\rho_0, c_0; 2\pi r/\lambda) \cdot u. \quad (2)$$

Here the constant of proportionality Z is the wave impedance, a function of fluid density ρ and speed of sound c , and, in general, of position r and the wavelength λ . For example, near a localized source the curvature of the wavefront causes the impedance to be complex, involving both a "resistive" real part (which represents the transport of acoustic power), and an inertial imaginary part (which represents localized or "near-field" effects). Very close to a small source, i.e., for small values of $2\pi r/\lambda$, the inertial impedance dominates. At a distance from the source, i.e., $2\pi r/\lambda > 1$, the resistive part becomes dominant. As wavefront curvature becomes smaller, i.e., $2\pi r/\lambda \gg 1$, the wave becomes essentially locally plane; and we have the plane-wave result

$$p = \rho_0 c_0 \cdot u, \quad (3)$$

where $\rho_0 c_0$ is called the characteristic impedance of the fluid.

Let us pause momentarily to explore some of the magnitudes of these acoustical quantities for familiar situations in air and then in water. We note that acoustical waves usually involve pressures and particle motions that are very small. For example, in conversation at a distance of 1 meter from the speaker, peak acoustic pressures are about

$$p \doteq 10^{-6} \text{ atmosphere} \doteq 1 \text{ dyne/cm}^2 \doteq 1.5 \times 10^{-5} \text{ psi.}$$

Correspondingly, we find

$$u \doteq \frac{1}{40} \text{ cm/sec} ,$$

and at a frequency of 1 kHz this represents a particle displacement

$$\xi \doteq 4 \times 10^{-6} \text{ cm} = 400 \text{ Angstroms.}$$

Some pertinent acoustical properties of air and water are compared in Table 1. These data show us that water is about 800 times denser than air, and about 20,000 times stiffer than air.* As a result we have for relative wave speeds and characteristic impedances:

$$c_{\text{water}} \doteq 5 c_{\text{air}}$$

$$\rho_0 c_0 (\text{water}) \doteq 4000 \rho_0 c_0 (\text{air}) \quad .$$

Thus for the same acoustic pressure considered above ($p = 1$ dyne/cm²), in water** we would find for particle velocity

$$u \doteq \frac{1}{160,000} \text{ cm/sec} \doteq 6 \times 10^{-6} \text{ cm/sec}$$

and for particle displacement at 1 kHz

$$\xi \doteq 1.0 \times 10^{-9} \text{ cm} = 0.1 \text{ Angstrom.}$$

These are indeed very small quantities! At a frequency of 1 kHz (about the frequency of the tone C''', two octaves above middle C) we would find for the acoustic wavelength

$$c_{\text{air}} \doteq 1.1 \text{ ft} \doteq 0.3 \text{ m}$$

$$c_{\text{water}} \doteq 5 \text{ ft} \doteq 1.5 \text{ m} .$$

* We have used the expression that bulk modulus is given by $\rho_0 c_0$.

** In a high-noise location -- namely, at the entrance to New York Harbor, during the daytime -- the rms acoustic pressure in a 1/3 octave band centered on 1 kHz is about 1 dyne/cm². (Based on data in Reference 1, p. 169).

The acoustic wavelength is a significant metric in determining whether a given object that radiates or interacts with a curve is "large" or "small". For any frequency, the acoustic wavelength in water is about five times that in air.

In the propagation of waves the "rules of the game" are as follows: The wave continues to propagate undisturbed as long as the properties of the medium remain unchanged. However, if there is a change in the properties, the wave is perturbed; and reflection (or scattering) results. When a plane wave strikes a plane surface at normal incidence (perpendicularly) a reflection takes place if the surface impedance Z_s (the ratio of acoustic pressure to acoustic particle velocity on the surface) is other than $\rho_0 c_0$. The situation is as shown in Figure 2; and the reflection ratio is given simply as (2)

$$\frac{p_r}{p_i} = \frac{Z_s - \rho_0 c_0}{Z_s + \rho_0 c_0} = \frac{z_s - 1}{z_s + 1}$$

where

p_i = pressure in incident wave

p_r = pressure in reflected wave

and

$$z_s \equiv Z_s / \rho_0 c_0$$

The corresponding "reflection loss" RL is

$$RL = -10 \log \left| \frac{p_r}{p_i} \right|^2 \text{ (dB)} \quad (5)$$

This is the logarithmic measure of the reflected intensity, relative to the incident value, and thus is a measure of the absorbing effectiveness of the surface.

Inspection of the above relation for p_r/p_i shows that when the surface impedance is equal to $\rho_0 c_0$, i.e., is "matched", there is perfect absorption (more about this later). On the other hand for a rigid surface $z_s \rightarrow \infty$, we find $p_r/p_i \rightarrow 1$; there is total reflection and the pressure at the surface, $p_r + p_i$ becomes $2p_i$, i.e., incident pressure is doubled. Conversely, if the surface impedance is very small, $z_s \rightarrow 0$, the result is $p_r/p_i \rightarrow -1$. Again this

corresponds to total reflection, but the pressure at the surface tends toward zero ($p_i + p_r \rightarrow 0$). Such a low-impedance surface is called "pressure release". A common example is a water-air interface, where waves incident from the water find essentially a zero-impedance surface. (Of course waves incident on the water from the air find a very high impedance surface).

In all of this discussion we must remember that impedance is a two-dimensional (complex) number. In general it involves both a real part or resistance, and an imaginary reactive part, which corresponds to either stiffness or mass. A graphical representation of the reflection arithmetic in terms of impedance is helpful in visualizing the mathematics of reflection. Figure 3 shows an impedance plane where real and imaginary parts of $z_s = z_s/\rho_0 c_0$ are represented by the horizontal and vertical axes, respectively. The circular contours shown represent loci of constant reflection loss RL. For example, the contour for RL = 10 dB indicates that a 10 dB reflection loss is provided by real values of z_s of 0.519 and 1.92, as well as any of the infinitely many complex values of z_s along the upper and lower arcs of the circle. From a presentation such as that of Figure 3 it is clear that matching a wave impedance well enough to achieve N dB reflection loss is equivalent to choosing values of z_s that lie in the "bullseye" defined by the circular contour RL=N dB. Of course hitting this bulls-eye must be arranged at the desired design conditions (frequency, temperature, etc.).

What kind of curve of RL vs frequency should we expect for a given treatment? Since z_s is in most cases a quantity that varies with frequency, we can visualize that the successive values can be plotted on a z_s plane like that of Figure 3. The contour generated by joining these values defines the values of RL vs frequency. A qualitative example is sketched in Figure 4. As frequency increases z_s varies, and RL increases to a maximum near frequency 3 and then decreases. It is helpful to note that the maximum in RL is defined by the closest approach to the point $z_s = 1$, and not necessarily by a particular "system resonance". We mention this because peaks in the frequency dependence of a particular quantity are customarily intuitively associated with simple resonance phenomena. Although such is sometimes the case with RL vs frequency, it is not necessarily so, especially with more complex absorbing systems where several local maxima are found. The frequencies at which maxima are obtained can shift with environmental conditions as the impedance contour is displaced on the z_s plane.

The above discussion is adequate for the normal-incidence situation. However, it is often important to provide absorption of waves at other angles of incidence.

It is easy to show that the "internal" impedance of a plane wave incident at an angle θ from the normal is $\rho_0 c_0 / \cos \theta$. Thus if, as is often the case, the absorbing surface presents an impedance that is "point reacting" (i.e., independent of incidence angle)*, the proper impedance for use in Equation (4) above for the reflection ratio is

$$z'_s = \frac{z_s}{\rho_0 c_0 / \cos \theta} \quad (6)$$

With this change the expressions and charts that are appropriate for normal incidence can be applied for incidence at other angles. The effect of varying angle can be significant. Consider, for example, a surface impedance that is matched at normal incidence, i.e., $z_s = \rho_0 c_0$; $z_s = 1$. For this surface there would be no reflection at normal incidence; but at off-normal incidence the impedance match is no longer perfect, and significant reflections occur. The situation is summarized in Table 2, which shows angle θ together with $z_s \cdot |p_r/p_i|$, and RL. For comparison we show in Table 3 the corresponding reflection parameters for a surface that is impedance matched for waves incident at 60° off normal, i.e., $z_s/\rho_0 c_0 = 2$.

If we compare the performance of the two surface impedances of Tables 2 and 3, we find an interesting and practical result. When we average intensity reflection ratio (as $|p_r/p_i|^2$) itself over angle of incidence from 0° to 75° we find that the two surfaces reflect roughly comparably, with the higher-impedance surface reflecting about 19 percent less than the lower. We note that this simple averaging over incidence angles represents the case in which the incident waves are uniformly distributed in a plane. (For example, waves incident on a wall from all angles in a horizontal plane.)

On the other hand if our concern is for a diffuse sound field with incident waves uniformly distributed over a hemisphere, we must weight the intensity reflection coefficients by $\sin \theta$ to account for the fact that there is

* The term "point reacting" relates to the fact that for such surfaces the local acoustic response velocity is proportional to the local acoustic pressure generated by the incident wave. This local response does not depend on the excitation and response at other points on the surface, at least for excitation wavelengths associated with incident acoustic waves.

more solid angle -- hence more contribution to intensity -- at angles away from normal. In this case the surface that is matched for $\theta = 0$ ($Z_s = \rho_0 c_0$) reflects about 2.5 times as much as does the surface matched for 60° ($Z_s = 2\rho_0 c_0$). This observation can be generalized. Acoustical materials such as ceiling tiles that are intended to absorb best for random (diffuse) incidence are designed with a surface impedance greater than $\rho_0 c_0$, as the above results indicate.

3. GRADUAL TRANSITION ABSORBERS

Where truly broadband and effective absorption is required, as in anechoic (echo-free) chambers for acoustical testing, it has been shown that a gradual transition in acoustic impedance can provide very good performance. As the term "gradual" implies, a certain amount of treatment thickness is required; a treatment of this type cannot be very thin relative to an acoustic wavelength. The purpose of such an impedance gradient is to minimize the wave reflection from the transition, i.e., to maximize the acceptance of the incident energy into the treatment.* The form of a graded-impedance treatment can be a truly gradual impedance change, or a number of layers of material with progressive impedance changes from $\rho_0 c_0$. An alternative to these continuous-surface treatments is an array of wedges. With such wedges the incident wave encounters a progressively increasing cross section of dissipative material, and the average impedance of a surface varies continuously with cross section. An installation of absorbing wedges in a large anechoic chamber is shown in Figure 5.

By careful design wedges can reduce pressure reflections to 10 percent or less at frequencies above that for which the wedge is $1/4$ acoustic wavelength in length. The pressure-reflection ratio measurements shown in Figure 6 illustrate the performance attained some years ago. Note that the "Harvard Wedge" by Beranek and Sleeper (ca. 1943) is longer than the "Berlin Wedge" by Meyer et al. (1940), and hence has a lower cut-off frequency. The total effective length of the Harvard wedge includes a backing air space of about 11 inches, which serves to improve the low-frequency

* From the standpoint of function it does not matter whether the energy is absorbed progressively as it travels through the transition, or is transmitted to a dissipative region beyond a transition region.

absorption without sacrificing performance at higher frequencies.* Clearly there is a materials saving in this approach. The same "trick" of introducing an air space between a treatment and a hard surface in order to improve the low-frequency performance is also utilized routinely in installations of acoustic tiles and other absorbing treatments.

Incidentally, the wedge configuration is also effective for the absorption of electromagnetic waves. For that application the absorption process requires an appropriate electrical resistivity instead of the acoustical "resistivity" provided by porous materials. Some multi-purpose anechoic chambers have been constructed wherein the acoustical wedges have been impregnated with graphite to provide the required absorption for electromagnetic waves. (3)

In Figure 7 there is shown a wedge-like installation as a lining for a water-filled test tank, together with the measured pressure-reflection coefficient. (4) An absorbing rubber was used to make the flat, two-dimensional wedges in varying sizes. The longest wedges (20 cm) are about 1 wavelength long at 7.5 kHz, which is roughly the lower cut-off frequency for 10 percent pressure reflection. Thus, this treatment is less effective than a full installation of 3-dimensional wedges, but it uses much less material. This obviously was a consideration in the design.

4. DISTRIBUTED ABSORPTION

In what has been presented above we have considered the absorption (i.e., nonreflection) of waves at a surface. We considered the impedance conditions required for absorption, with the implication that the waves were not returned from within the surface. Obviously the best "absorber" of acoustic waves in a fluid is simply an unbounded region of the same fluid. The absorption would be perfect at any

* The wedges discussed here are designed for installation in front of a rigid wall. Since these absorbers for airborne sound dissipate energy through viscous motion of the air within the fibrous structure of the wedge, and since the rigid wall prevents air particle motion in its immediate vicinity (say, a small fraction of an acoustic wavelength), one may omit the fibrous structure from a region next to the wall with very little performance penalty in the low-frequency region.

angle of incidence (such a fluid half-space is not point-reacting). The waves would never be reflected, and no dissipation would be required. However, in the case of an acoustical treatment of finite thickness the acoustical energy that enters the treatment must be dissipated. What are the dissipation mechanisms that make it possible to absorb the energy in a wave that is incident on a surface? In what follows we shall review mechanisms for dissipation and some of the important implications for the design and performance of absorption systems.

Mechanisms of Absorption

Any of a variety of energy loss mechanisms can account for distributed dissipation. Some examples are:

- viscous flow in fibrous materials
- viscoelasticity
- conversion (removal) of wave energy to waves of a different direction or type (this will be discussed later)
- thermal losses, e.g., the conduction of heat to a fibrous structure within which waves are propagating, heat losses from bubbles in a liquid, etc.
- molecular relaxation, e.g., CO₂ in air.

For absorbing systems in air the dissipation is most commonly accomplished within fibrous materials. There the acoustically induced air motions dissipate energy through air viscosity in the resulting shear flows. This phenomenon allows the tailoring of the structure via fiber size and orientation, and via the packing or density of the fibers. Surface layers representing flow resistance, mass, or both are also used in absorbers for airborne sound. The flow resistance of such a layer can provide a significant part of the treatment's dissipation. Alternatively, the mass of such a surface layer can act with the compliance of the air space behind it to provide a resonant absorption peak in a desired frequency range. (Absorption is provided by fibrous material in the air space.) As an alternative, when a finished (e.g., painted) surface is desired, holes or fissures can be provided to allow the generation of acoustic flow within the fibrous interior of a treatment.

In underwater sound applications absorbers are of interest, for example, as liners for acoustical test tanks or spaces, and for providing proper conditions for transducers. Some treatments have used the mechanism of

viscous flow through screens. (5) (In some cases the screens were surrounded by viscous oil, and were separated from the acoustical space by a light membrane.) However, because the characteristic impedance of water is so large, it is difficult to keep a screen from moving with the wave. (Such motion would minimize the differential motion between the screen and the viscous oil, which motion is necessary to produce losses.) Recall that the characteristic impedance of water is about 4000 times that of air. Thus, as regards the mass reactance of a sheet or layer (as measured by the dimensionless ratio $2\omega m/\rho_0 c_0$), and hence its transmission and reflection properties, a sheet of paper in air is equivalent to about a 1-inch steel plate in water! Therefore, a light screen would present very little inertia to an acoustic wave in water, and would be forced to move with the wave.

An absorption approach that has proven more successful for underwater sound than viscous flow through screens has been the use of viscoelastic materials with internal voids. In such materials some of the 'acoustical compressional motion is converted to shear deformation by the voids. The shear viscoelasticity then results in dissipation. Dissipative materials of this type can be designed to provide the desired surface impedance, at least over limited ranges of frequency and environmental parameters.

Physical Effects of Dissipation

Any type of dissipation in a wave-bearing medium affects a travelling wave by withdrawing energy from it. In Figure 8 there is sketched a wave travelling in the positive x-direction. For illustrative purposes we say that the medium exhibits a type of hysteresis, as indicated at the top of the Figure. Stress and strain are not in phase, and thus during each cycle energy in proportion to the area of the hysteresis loop is dissipated. The effect of such hysteresis is to remove energy from the wave, reducing its amplitude as it travels.

If the local rate of energy dissipation is proportional to the local energy density (e.g., in a system with linear viscoelasticity), then the temporal decay of the energy E is exponential:

$$E = E_0 e^{-\omega\eta t} \quad (7)$$

Here E_0 = energy at time $t = 0$

ω = $2\pi f$, circular frequency

η = loss factor (see below).

The loss factor defined by the above general relation is a most useful measure of dissipation. If we write $\Pi_{\text{diss}} = -dE/dt$ for the power dissipation, which accounts for the rate of decrease of energy, we have

$$\eta = \frac{\Pi_{\text{diss}}}{2\pi f E} \quad (8)$$

The resulting spatial attenuation rate for the travelling wave can be shown to be

$$\Delta_\lambda = 27.3\eta \frac{c}{c_g} \text{ (dB/wavelength)} \quad (9)$$

where

c_g = phase of the wave

c = "group" or energy speed of the wave.*

The ratio of these speeds must be introduced in general because the wavelength is governed by the phase speed, while energy travels at the group speed. For longitudinal waves in a fluid we may say $c/c_g = 1$; but for some other wavebearing systems this is not so. An important case that will be considered shortly is that of plate flexural waves, for which $c/c_g \doteq 0.5$.

The implications of the above relationship become more clear when we consider typical loss factors of materials of interest. Consider simple longitudinal or shear wave ($c/c_g \doteq 1$) for which

$$\Delta_\lambda \doteq 27\eta \text{ (dB/wavelength)}.$$

For structural materials (e.g., metals) the loss factor η is typically very small, say $\eta \sim 10^{-3}$. This corresponds to attenuation $\Delta_\lambda \doteq 0.03$ (dB/ λ) that is more-or-less negligible for most engineering purposes. In strong contrast, however, is the case of viscoelastic materials, for many of which the loss factor lies in the range $0.1 < \eta < 1.0$. Corresponding

* This is the speed of the envelope of a wave "packet", i.e., the speed at which physically realizable linear combinations of waves with finite bandwidth propagate. It differs from the phase speed in dispersive media, i.e., media in which wave speed is frequency-dependent.

attenuations of 2.7 to 27 dB/wavelength are indeed significant. (We must note that these loss factors apply to the elastic [Young's] and shear moduli, and therefore to waves governed by them. Bulk waves are controlled by the bulk modulus for which the loss factor is generally much lower.)

Let us explore the question of when a treatment with internal dissipation is "thick enough". Earlier we considered the requirement for impedance matching so that incident energy will enter an absorbing treatment with a known (small) reflection at the front surface. Of course, in the absence of dissipation in a treatment of finite thickness, the accepted energy will be reflected from the inner surface, returning again to the front surface to defeat the purpose of the careful matching of impedance.

However, with some dissipation (as there always is to some degree), at higher frequencies the reflection from the back side of a layer eventually becomes unimportant, and the input properties are indistinguishable from those of a layer of infinite thickness. We require a measure of the round-trip attenuation of a wave; and it follows from the preceding discussion that, since the loss factor η (corresponding to distributed dissipation) defines attenuation per wavelength, the appropriate measure for a layer of thickness h involves h/λ (the number of wavelengths) and η . Specifically a useful rule of thumb is that a "thick" (very thick) layer in this context is one for which

$$kh\eta = 2\pi h\eta/\lambda > 1.$$

This inequality implies an attenuation of at least 4.3 dB each way through the layer (a total of 8.6 dB). In such a case the two-way-transmitted wave is reduced in intensity of less than 1/7 of its initial value, and for many purposes this represents an insignificant reflection component. The attenuation increases rapidly with frequency or with loss factor. For example, for $kh\eta = 2$, the relative intensity of the two-way-transmitted wave drops to less than 0.02, even for perfect reflection at the back surface of the layer.

The above measure is helpful in assessing the thickness of a layer; but it might wrongly suggest that one could provide a highly effective absorbing treatment simply by choosing the right characteristic impedance ρc for the material, and including enough dissipation via loss factor η so that the layer appeared infinitely thick. Such a simplified view is misleading. Consider the perfect impedance match required for absorbing waves in a lossless fluid. The impedance should be equal to $\rho_0 c_0$ for the fluid, i.e., a real number. (The best possible absorber would be

simply more of the same fluid.) But in general when a medium is made dissipative, its characteristic impedance ceases to be a real number, and becomes complex. Thus, it is probably not possible to make a near perfect absorber by using a single layer of a highly absorptive material of only modest thickness, since there will always be some reflection generated by the impedance mismatch at the surface.

As an example consider a viscoelastic layer (with internal voids) as an absorber of waterborne sound. The characteristic impedance is

$$\rho c = \sqrt{\rho E(1-i\eta)} \quad ,$$

where E is the effective modulus of the layer. Since this impedance is a complex number, a perfect match to the impedance of water cannot be achieved. The presence of dissipation introduces an inertial or massive impedance. (We use a time dependence $e^{-i\omega t}$ here.)

For such a dissipative layer intended to produce a reflection loss $RL = 20$ dB, i.e., a pressure reflection coefficient of 0.1, not only must the impedance magnitude be matched to that of the water, but the loss factor must be less than about 0.5. (Table 4 gives values of maximum RL for a range of loss factors.) How thick must such a layer be? Let us say that the two-way-transmitted wave must be about 10 dB below the level of the front-surface reflected wave, so as to contribute only minimally. This puts the two way transmission at about $RL + 10 = 30$ dB. Given that $\eta \approx 0.5$, we expect attenuation of $27 \eta \approx 14$ dB/wavelength. Thus, if we assume that the wave speed in the layer is approximately the same as that in water, the layer must be somewhat greater than one acoustic wavelength in thickness. This would imply a thickness of about 5 ft at 1 kHz, or 1 ft at 5 kHz! Obviously, it is not possible for effective absorbers at these frequencies to be of insignificant thickness. A magical paint will not do the job! It is interesting to compare the last example above with the observations made earlier on wedge type absorbers. In air, well designed wedges can provide 20 dB reflection loss for a wedge length of about $1/4$ acoustic wavelength (at the low-frequency limit of performance). On the other hand, the single dissipative layer concept considered above for waterborne sound appears to require more nearly a full wavelength in thickness. Incidentally, the configuration of flat, sheet-like (underwater) wedges shown in Figure 7 also required a treatment thickness of about one acoustic wavelength. These however should not have been expected to perform as effectively as the 3-dimensional wedges used for airborne sound. Two tentative conclusions are suggested:

- 1) absorbing treatments involving a gradual transition in acoustic properties (including wedges) appear more effective than a single dissipative layer, and
- 2) given present-day technology it appears more difficult to make broadband absorbers for underwater sound than for airborne sound.

5. SCATTERING

In many cases one may extend the concept of absorption of energy from an acoustic wave. We have regarded absorption as the conversion of the wave energy to heat, thus removing the energy from the acoustical system of concern. Equally valid, however, is the removal of energy from the wave by conversion to another wave system, another direction of propagation, or another wave type. The energy so removed would eventually be converted to heat, but it is useful to recognize the intermediate step.

Thus, the redirection of the energy of an incident wave may be interpreted as absorption if it prevents the energy from returning coherently to the observer. There are techniques for accomplishing this by contouring a surface e.g., large convex scattering surfaces on the side walls of an auditorium that serve to suppress discrete echos and to diffuse the sound. Direct reflections can also be reduced by treating a surface with closely spaced areas of alternating high and low impedance. Such a surface can, in principle, reduce direct specular reflection by interference between reflections from the disparate surface regions. The acoustical energy that appears to be absorbed actually is scattered in nonspecular directions. Such an arrangement can of course generate significant backscatter (i.e., reflection) at off-normal incidence at higher frequencies where the impedance regions are no longer small relative to the acoustic wavelength.

In the context of distributed dissipation, scattering can also play a role. Acoustic waves will be scattered from any local variation in the properties of the medium. As is sketched in Figure 9, a small local discontinuity in compressibility will result in a localized volume source that radiates uniformly in all directions. In contrast, a small local discontinuity in density generates a localized force in response to an incident wave, generating an acoustic-dipole radiation aligned with the direction of travel of the incident wave. In addition, if the medium can support shear waves, as in an elastic solid, the localized force will also generate shear waves as indicated.

For such a small localized discontinuity both the monopole and dipole acoustic radiation terms are proportional to $f^4 a^6$ where f is frequency and a is the radius of the discontinuity. Thus the scattering increases strongly with both particle size and frequency. We might remind ourselves that stronger scattering at higher frequencies (small wavelengths) is a familiar concept. It accounts for the predominately blue light scattered in the atmosphere, and the predominately red color of the setting sun (i.e., that which is left over after the scattering of the blue light).

Resonant Scatterers/Absorbers

Up to now we have considered only plane acoustic waves for which the wave impedance is a real number, i.e., a resistance $\rho_0 c_0$. On the other hand, when there is significant curvature in a wave front, the local wave impedance contains an inertial term. For small objects that pulsate or oscillate, like the small scatterers discussed above, the acoustic impedance is predominantly inertial (and is of magnitude smaller than $\rho_0 c_0$). Thus, if a compliance is associated with the deformation of the scatterer, there is the possibility of a resonance with the inertial acoustic load.

An example of such a resonant scatterer is a bubble in a liquid. For a small air bubble of diameter d cm in water at a pressure of N atmospheres the resonance frequency is (1,6)

$$f = \frac{0.656 \sqrt{N}}{d} \quad (\text{kHz}) \quad . \quad (10)$$

For example a 1/8 in. diameter bubble ($d \approx 0.3$ cm) is resonant just above 2 kHz at 1 atmosphere.

A resonant scatterer can be a strong scatterer. It can redirect a considerable amount of energy, and can also dissipate significantly if it incorporates an appropriate loss mechanism. An example of this latter effect is the use of foamed neoprene strips as resonant absorbers in an acoustic test tank. (7) These absorbers remove energy with an effectiveness of perfect absorption many times their surface i.e., they have a large "absorption cross section". Dissipation is due to the internal losses in the viscoelastic neoprene, and serves to broaden the bandwidth of the absorbers. It is not easy to see an analog of this phenomenon for use in absorbing airborne sound. (Bubbles in air?) However, resonant absorbers have been constructed (as Helmholtz or other cavity resonators) especially for producing desired absorption in specified frequency ranges. (8)

Functional Absorbers

Another type of absorber that is often useful is the "functional absorber". Such absorbers are panels, cubes, tetrahedrons, etc. of absorbing material designed to be distributed within a volume where absorption is required. These are broadband absorbers, and can be rendered more effective when placed (e.g., suspended) near bothersome noise sources such as machines in a manufacturing area. The absorbing behavior is somewhat dependent on their spacing and arrangement because they are large enough to influence the sound field in which they are placed.

6. STRUCTURAL DAMPING

The damping of flexural waves in structures deserves a place in our consideration of acoustical attenuation systems. With their principal displacements normal to a plate surface, flexural waves play an important part in the generation of and response to sound fields, both in air and in water. Plates and beams which are incorporated in the structures of vehicles, machines, appliances, and buildings can convert flexural-wave energy into sound. Damping is often useful in applications ranging from aesthetics and customer appeal, to noise reduction, hearing protection, and the inhibition of structural fatigue.

Here again a most useful measure of the degree of damping is the loss factor η defined as before

$$\eta = \frac{\Pi_{\text{diss}}}{2\pi f E} = \frac{E_{\text{diss}}}{2\pi E} \quad (11)$$

Here Π_{diss} is the power dissipated, f is the frequency, E_{diss} is the energy dissipated per cycle, and E is the stored energy of the vibration. Note: Π_{diss} and E may apply to a complete vibrating system, or to a typical unit area or unit length.

What are typical levels of loss factor? We find that values of η range from 10^{-3} - 10^{-4} for metals, to $(0.5 - 2) \times 10^{-2}$ for built-up structures fabricated from plates, beams, etc. without appreciable explicit damping. Loss factors in the range 0.05 - 0.2 are realized by careful design of damping treatments. Loss factors as high as 1.0 or occasionally higher are found for viscoelastic materials in the temperature and frequency region around their glass-transition temperatures. The above definition of the loss factor makes it clear that the appropriate measure of damping is the ratio of energy-dissipated-per cycle, E_{diss} , to the stored energy E . To achieve appreciable values of damping requires a dissipation mechanism, most often

involving a high loss material. However, a high-loss material (e.g., a viscoelastic material having a high loss factor) is not enough in itself. It must be incorporated in the vibrating system in such a way as to share significantly in the system energy. This means that a damping treatment cannot be flimsy and yet effective. (There can be some exceptions, say, in the case of resonant damping treatments, but the preceding statement is a good general guide.)

Free and Constrained Viscoelastic Layers

Two widely applied damping configurations that make use of viscoelastic materials are the free viscoelastic layer and the constrained viscoelastic layer. These are sketched in Figure 10 as applied to a plate or beam. As the plate is deformed in flexure, the free viscoelastic layer flexes with it, participating in the bending stiffness as a part of a two-layer beam. To allow this behavior the viscoelastic layer must be tightly bonded to the plate, and must be continuous, i.e., not subdivided by unbonded joints, over a distance at least as great as above $1/2$ flexural wavelength. The performance of a free viscoelastic layer in useful thicknesses is described approximately as follows (9):

$$\eta_{\text{composite}} \doteq \text{const.} \frac{E_2}{E_1} \left(\frac{H_2}{H_1} \right)^2 \eta_2 \quad (12)$$

Here E and H are the Young's modulus and thickness respectively, and subscripts 1 and 2 refer to the plate and the viscoelastic layer, respectively. The composite loss factor increases with the modulus and loss factor of the viscoelastic layer, and with the square of the layer thickness. Oberst(10) recognized this, and designed a series of filled elastomer damping materials with high (modulus)-(loss-factor) products. In addition, he observed that not only does the material not need to be heavy, but that it is probably better being light so that a given weight will mean a thicker, more effective treatment.

The behavior of a constrained viscoelastic layer, on the other hand, involves the shear deformation of the layer (11). (See Figure 10). Significant damping can be achieved in the frequency range (i.e., wavelength range) where there is a balance between the shear stiffness of the viscoelastic layer (2) and the extensional stiffness of the constraining

layer (3). In this region the maximum composite loss factor varies approximately as follows(9):

$$\eta_{\text{composite}} \doteq \text{const.} \frac{E_3 H_3}{E_1 H_1} \cdot \eta_2 \quad (13)$$

As is indicated in Figure 10, the subscripts 1,2,3 refer to the plate, the constrained viscoelastic layer, and the constraining layer respectively.

Thus, for a constrained-layer treatment good damping required that the constraining layer be extensionally stiff, and that the viscoelastic layer have a high loss factor. The modulus of the viscoelastic layer does not appear explicitly above, but it determines the layer thickness required to place the maximum damping in the desired frequency-temperature region.

Thickness-Resonant Damping Layer

A relatively thick viscoelastic layer can provide good damping at frequencies high enough for the layer to be resonant in its thickness direction. The configuration and its typical performance are sketched in Figure 11.(12) Maxima in damping occur where the layer thickness becomes $1/4$, $3/4$, etc. times the longitudinal wavelength in the thickness direction through the layer. That is, maximum damping is found at antiresonances of the layer, where the input impedance as seen by the plate is large. Note that this description of the system dynamics requires that the viscoelastic layer be essentially free, e.g., air backed.

This behavior is analogous to that of a dynamic absorber. There is a balance between the maximum damping and the frequency bandwidth as the material loss factor is varied. A bandwidth of two octaves with $\eta_{\text{composite}} > 0.1$ can be achieved in practice.

Such treatments tend to be heavy. In addition the viscoelastic material is likely to be too soft to be effective as a free layer at low frequencies, i.e., well below the resonant range. The layer thickness and wavespeed determine the frequency range of effective damping. This range will vary somewhat with temperature, because the layer modulus is temperature dependent.

Fluid Loaded Resonant Layer

When a plate damping treatment is to be water-backed, it is possible to use the loading impedance of the water as a part of the damping system. When the speed of flexural waves in the plate is less than the speed of sound in water (i.e., the frequency is below the "critical" frequency), the fluid impedance is nearly purely inertial.* This inertial water load can resonate on the lossy-compliant layer producing a large dissipative load on the plate. The behavior of the system is complicated by the dependence of both plate wavespeed and fluid load on frequency.

Typical behavior is shown in Figure 12. The damping can be quite large, and has a useful bandwidth. Because the water load provides a backing mass for the compliant layer, the system (layer) incremental weight and thickness can be less than those for comparable treatments of, say, the thickness-resonant type.

7. SUMMARY

If an acoustic wave is to be absorbed by a treatment of limited thickness, two things must happen: a) the acoustic impedance of the treatment must be adequately matched to that of the wave-bearing medium from which the wave is incident, so that the reflection will be minimized and the wave energy will enter the treatment; and b) an effective dissipation mechanism must be provided within the treatment so that the energy will be absorbed before it can be reflected within the treatment and retransmitted to the external medium.

Achieving absorption performance that is broadband in frequency requires a treatment thickness that is in some way comparable to the acoustic wavelength at the lowest frequency (i.e., largest wavelength) of interest. However, absorption over a limited bandwidth can be achieved with resonant systems that need not be as thick as broadband absorbers. Such resonant treatments are useful at lower frequencies or where performance bandwidth can be sacrificed in order to reduce the space occupied by the treatment.

* Note that water loads the plate by increasing its effective mass, and reducing the speed of flexural waves. The water load does not damp the plate appreciably in this frequency range.

Scattering can serve to remove energy from a wave, providing attenuation of the wave even though the actual dissipation of the energy as heat may occur as a secondary process. Scatterers may be either resonant or nonresonant.

The attenuation of acoustical waves in structures is of practical importance in noise control. Flexural waves play a major role in both transmission and radiation of acoustic energy. There is available a variety of damping treatments for attenuating flexural waves; included are viscoelastic layers deforming principally in extension or in shear, and several types of resonant treatments also involving viscoelastic materials.

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TABLE 1
ACOUSTICAL PARAMETERS OF AIR AND WATER (Approximate Values)

Air ($T=20^{\circ}\text{C}$, $P=1$ atm.)	Mks	cgs	English
Density ρ_o	1.21 (kg/m^3)	1.21×10^{-3} (gm/cm^3)	2.34 (slug/ft^3)
Speed of Sound c_o	343 (m/sec)	3.43×10^4 (cm/sec)	1127 (ft/sec)
Characteristic Impedance $\rho_o c_o$	413 ($\text{kg/m}^2\text{-sec}$)	41.3 ($\text{gm/cm}^2\text{-sec}$)	2.64 ($\text{slug/ft}^2\text{-sec}$)
<u>Water*</u>			
Density ρ_o	1000 (kg/m^3)	1.0 (gm/cm^3)	2.05 (slug/ft^3)
Speed of Sound c_o	1500 (m/sec)	1.5×10^5 (cm/sec)	4920 (ft/sec)
Characteristic Impedance $\rho_o c_o$	1.5×10^6 ($\text{kg/m}^2\text{-sec}$)	1.5×10^5 ($\text{gm/cm}^2\text{-sec}$)	1.0×10^4 ($\text{slug/ft}^2\text{-sec}$)

* These are representative values. Properties vary with temperature, pressure, and salinity.

TABLE 2
REFLECTION PARAMETERS FOR OFF-NORMAL INCIDENCE FOR
A SURFACE THAT IS IMPEDANCE MATCHED AT NORMAL INCIDENCE

θ (degrees)	$z_s = \frac{Z_s \cos \theta}{\rho_o c_o}$	$ p_r/p_i $	RL (dB)
0	1.0	0	∞
15	0.97	0.017	35
30	0.87	0.072	23
45	0.71	0.17	15
60	0.5	0.33	9.5
75	0.26	0.59	4.6

TABLE 3

REFLECTION PARAMETERS FOR OFF-NORMAL INCIDENCE
FOR A SURFACE THAT IS IMPEDANCE MATCHED AT 60° INCIDENCE

θ (degrees)	$z_s = \frac{Z_s \cos \theta}{\rho_o c_o}$	$ p_r/p_i $	RL (dB)
0	2	0.33	9.6
15	1.93	0.32	10.0
30	1.73	0.27	11.4
45	1.41	0.17	15.4
60	1.0	0	∞
75	0.52	0.32	10.0

TABLE 4

MAXIMUM REFLECTION LOSS RL_{\max} ACHIEVABLE FOR A VERY THICK ABSORBING LAYER HAVING LOSS FACTOR η . MAGNITUDE OF LAYER CHARACTERISTIC IMPEDANCE IS EQUAL TO $\rho_0 c_0$ OF THE FLUID AT MAXIMUM ABSORPTION.

<u>LOSS FACTOR η</u>	<u>RL_{\max} (dB)</u>
0.1	32
0.2	26
0.3	23
0.5	19
1.0	14

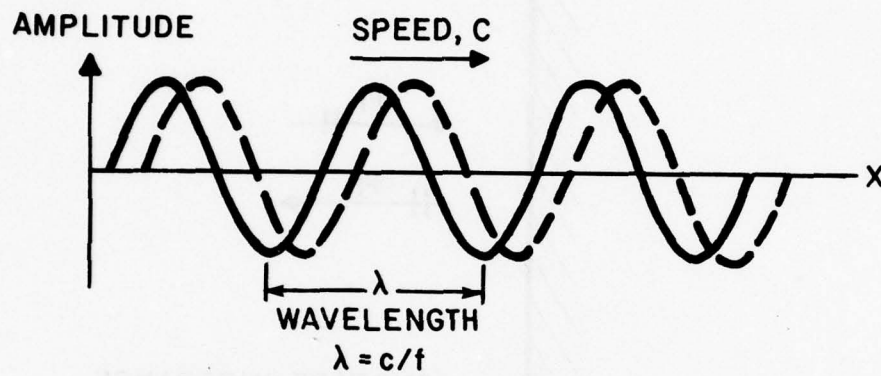


Fig. 1 A Traveling Wave Sketched at Two Successive Times

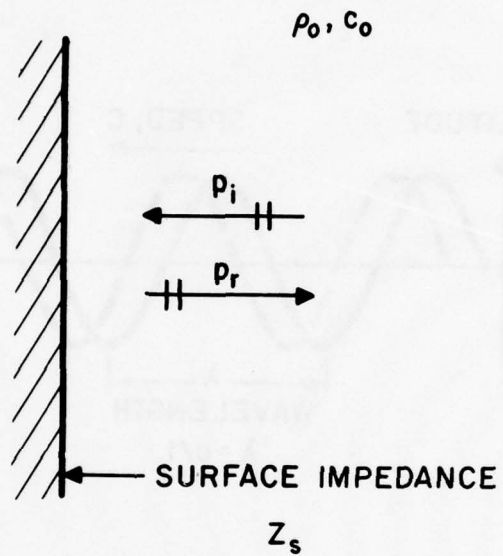


Fig. 2 Incident and Reflected Acoustic Pressures p_i and p_r for Normal Incidence of Plane Waves on a Plane Surface

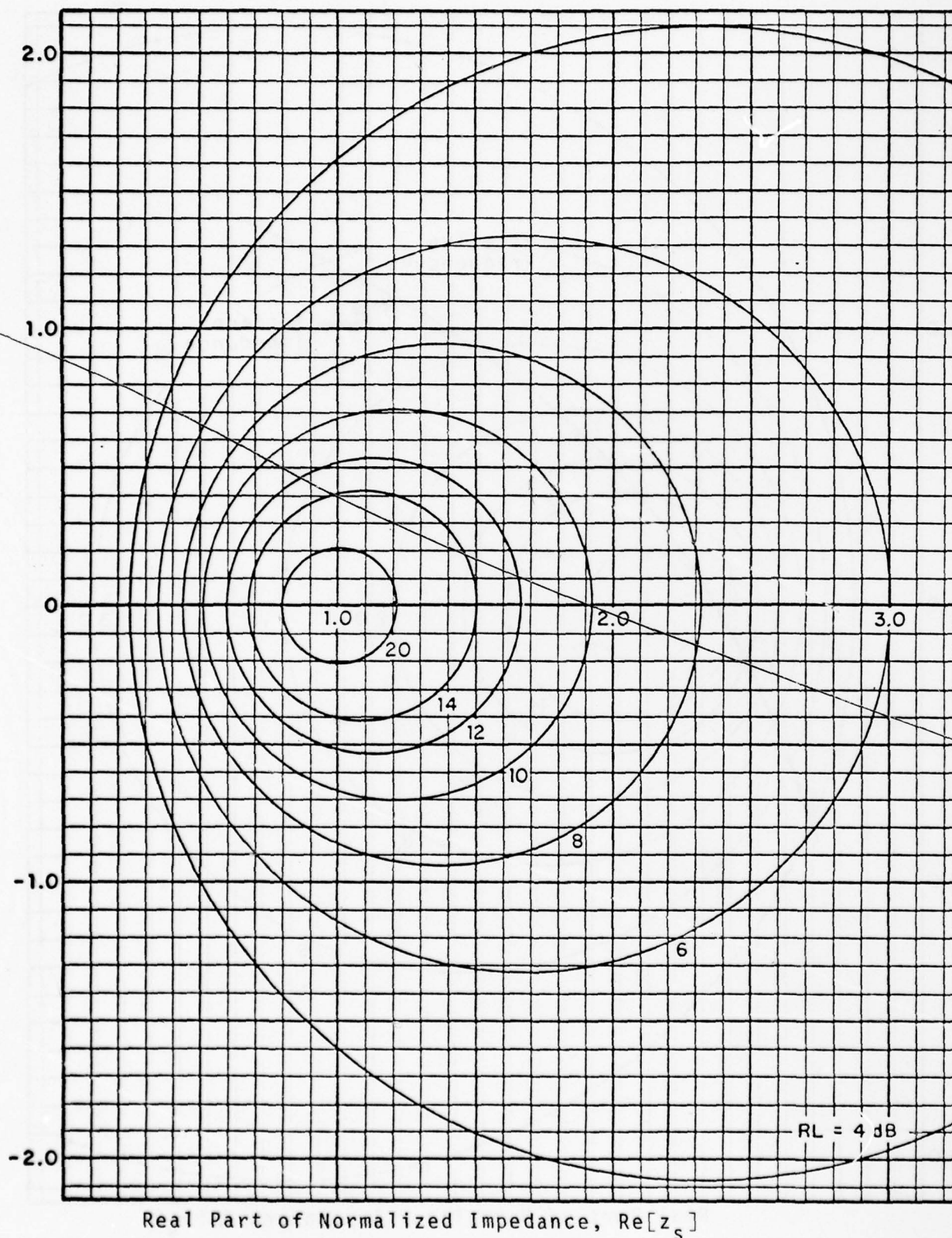


Fig. 3 Contours of Constant Reflection Loss RL for Plane Waves Incident on a Surface Having Complex Impedance Z_s . (Note that chart uses normalized impedance $z_s = Z_s / \rho_0 c_0^s$).

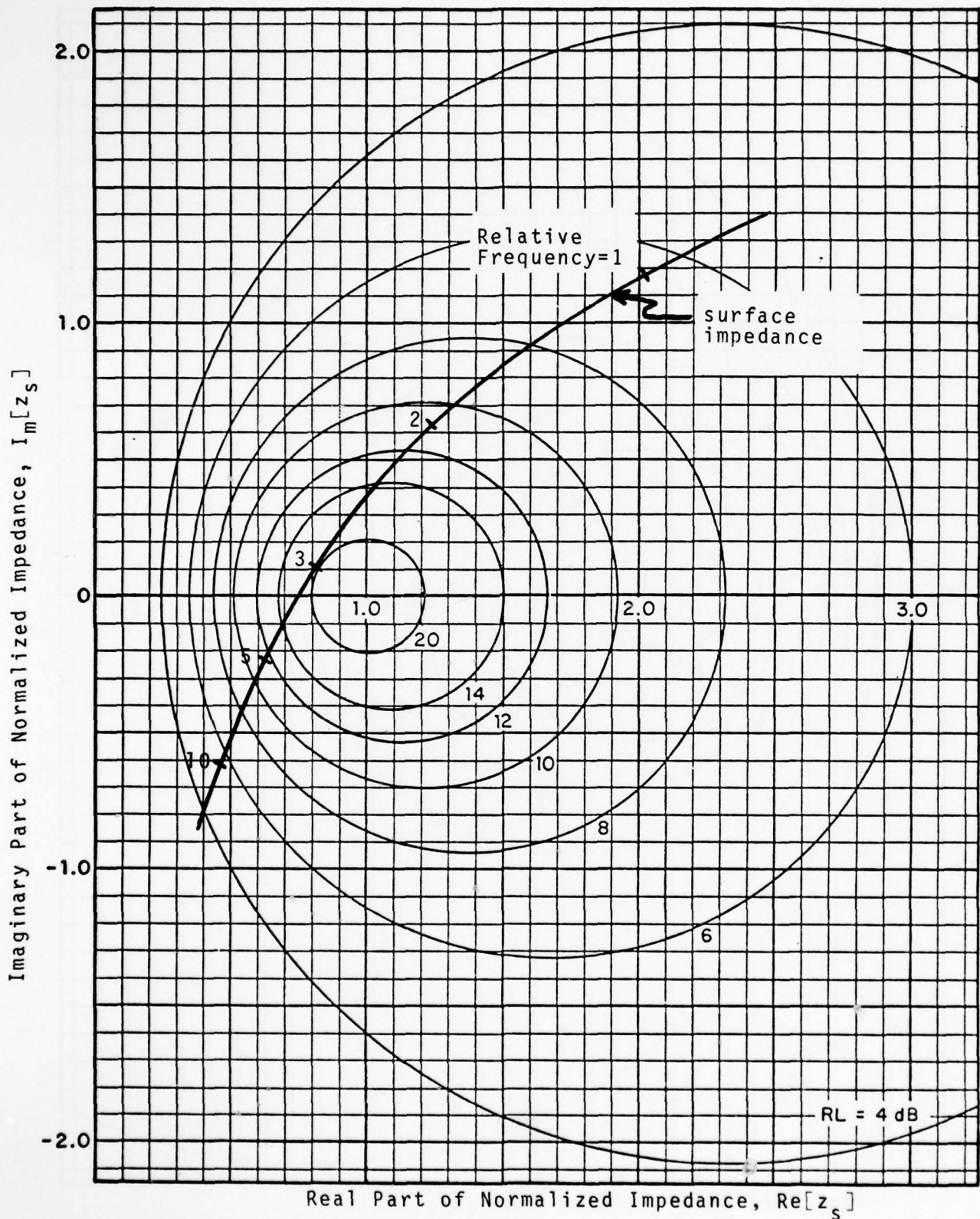


Fig. 4 A Hypothetical Example of an Impedance Contour for an Absorbing Surface.

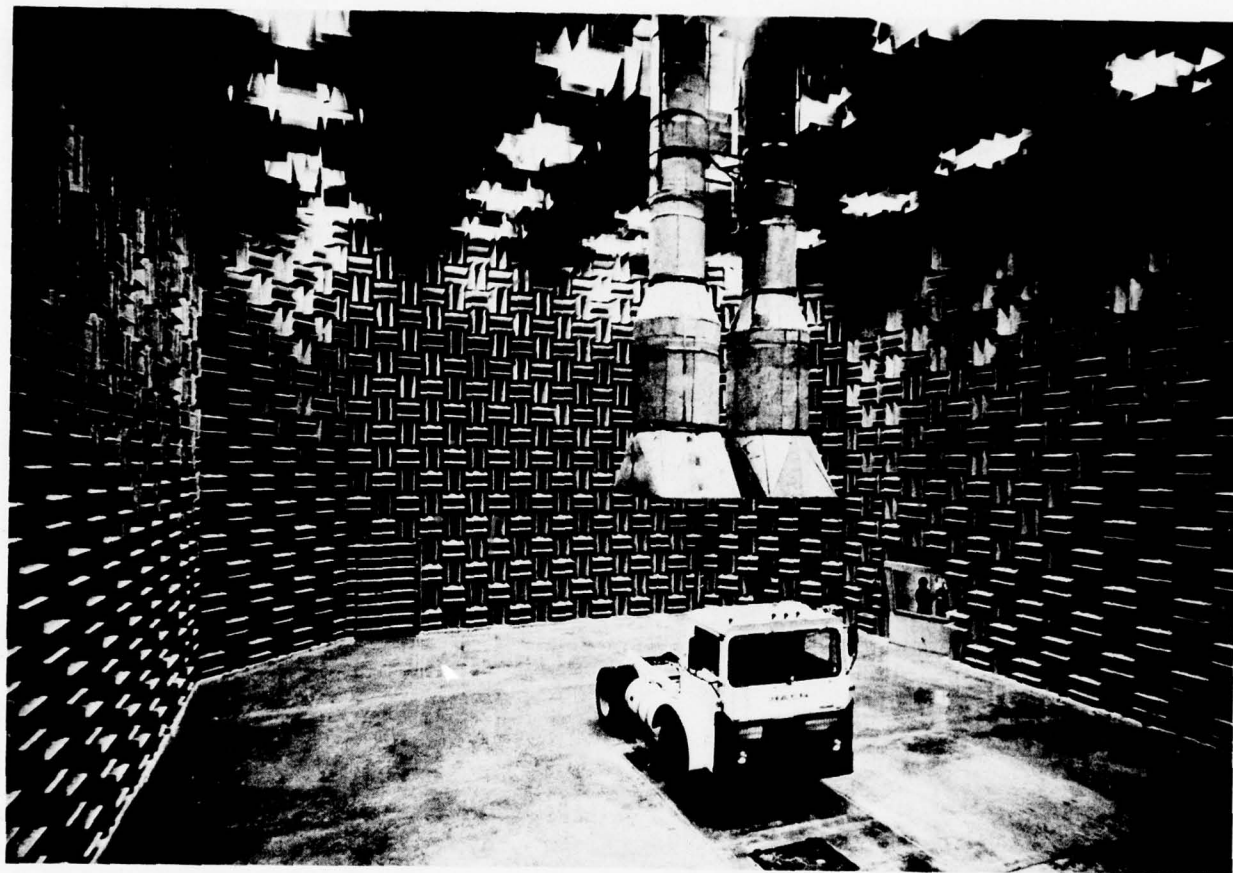
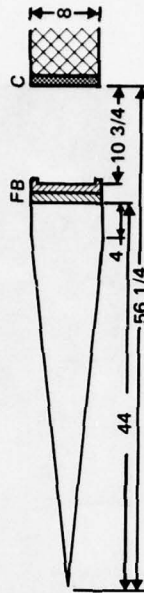
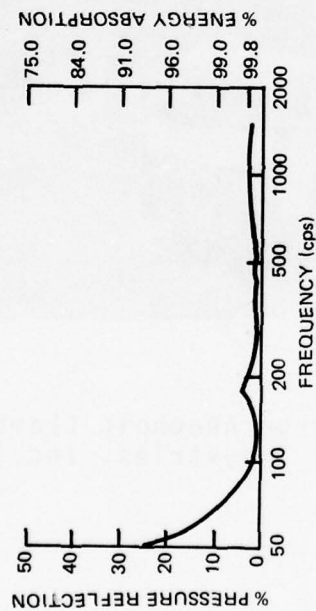


Fig. 5 Absorbing Wedges in a Large Anechoic Chamber.
(Photo Courtesy of Eckel Industries, Inc.)

Harvard Linear Wedge Structure
Structure No. A

- A. 4" PF Fibreglas - 2.5 lb/ft³; Rf-30 g cm⁻² sec⁻¹
 B. 2" PF Fibreglas Blanket - 3.25 lb/ft³
 C. 3/4" Wooden Frame
 D. 1" Cork
 E. Muslin Bag - 0.016 lb/ft²; Rf-1 g cm⁻² sec⁻¹



Meyer, Buchmann, and Schoch-Berlin
Pyramidal Structure
Structure No. B

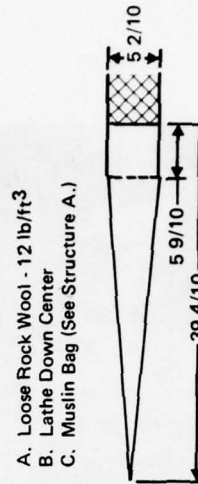
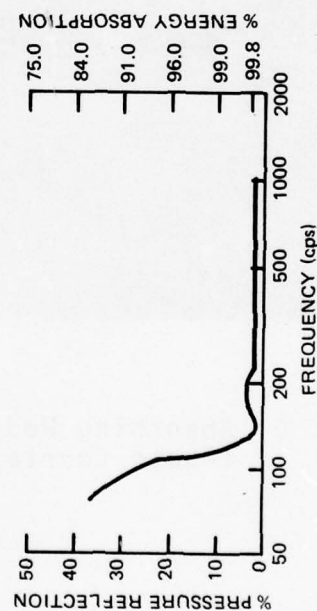


Fig. 6 Pressure-Reflection Coefficients for Two Absorbing
Wedges in Air ("Normal" Incidence, i.e., Along Long
Axis of Wedge) [After Beranek and Sleeper, J. Acoust.
Soc. Am., 18, 140-150 (1946)]

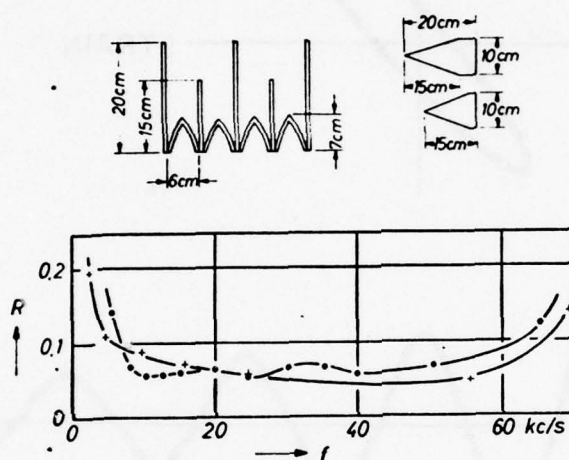


Fig. 7 Pressure-Reflection Coefficient for "Rib Absorbers" for Underwater Sound (after Meyer, Ref. 4).

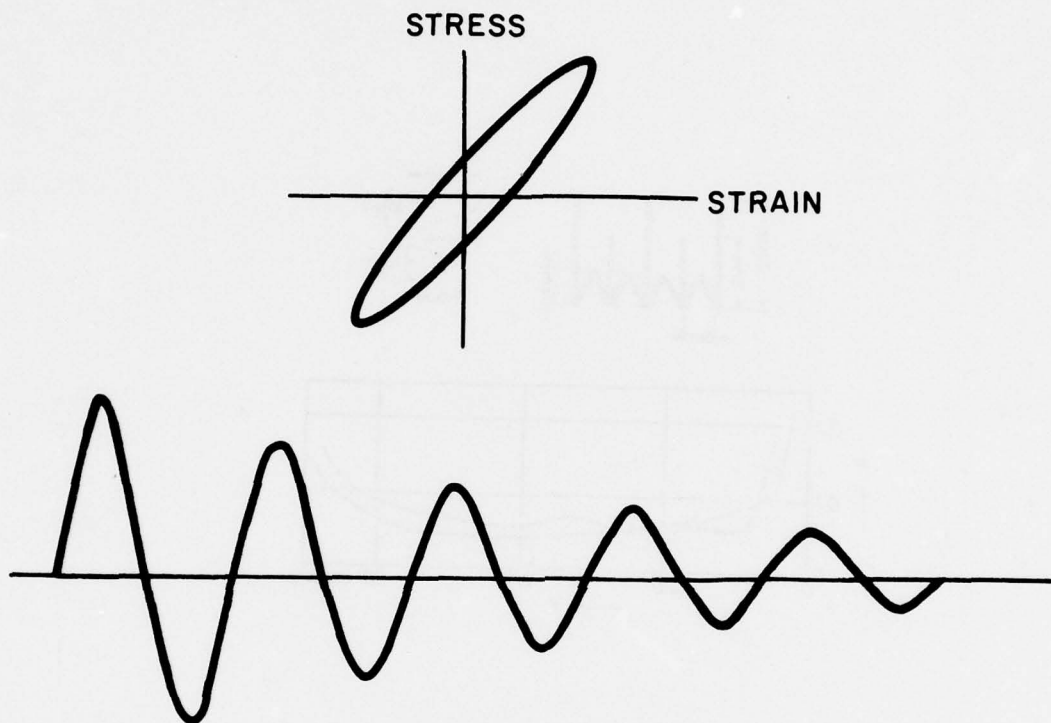
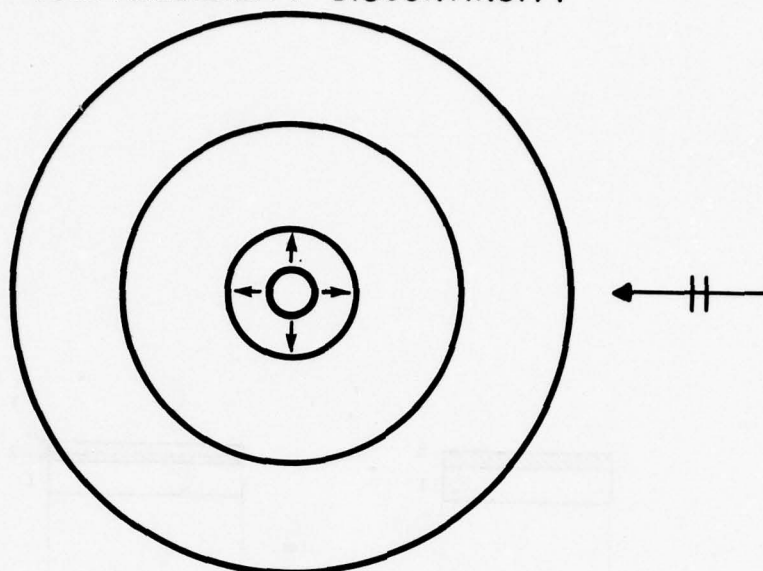


Fig. 8 Traveling Wave with Attenuation Due to Hysteretic Losses

COMPRESSIBILITY DISCONTINUITY



DENSITY DISCONTINUITY

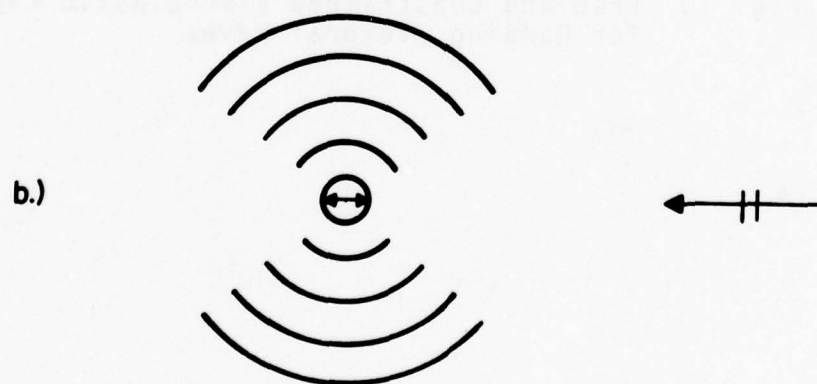
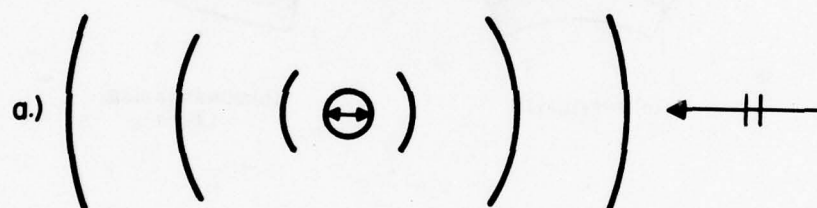


Fig. 9 Scattering from localized discontinuities. A Compressibility discontinuity generates omnidirectional monopole acoustic radiation. A density discontinuity generates directive dipole acoustic radiation (a), and also — in an elastic solid — shear waves (b).

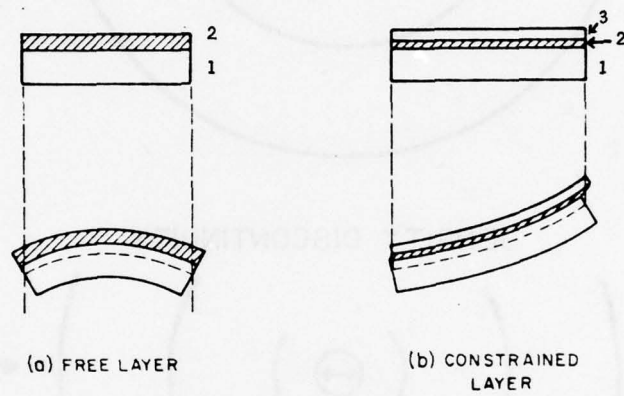


Fig. 10 Free and Constrained Viscoelastic Layers for Damping Flexural Waves

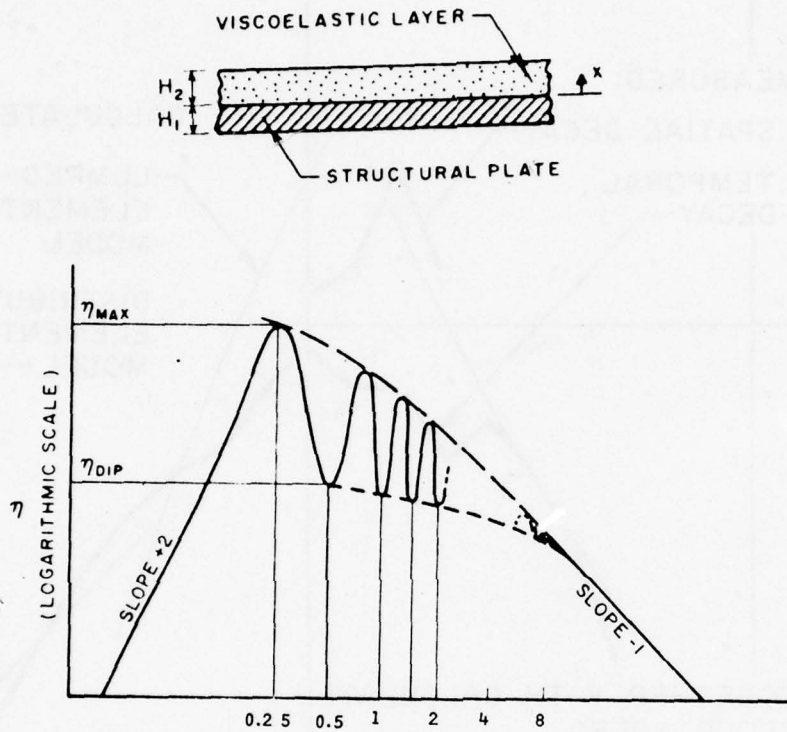


Fig. 11 Typical Expected Performance of Thickness-Resonant Viscoelastic Damping Layer

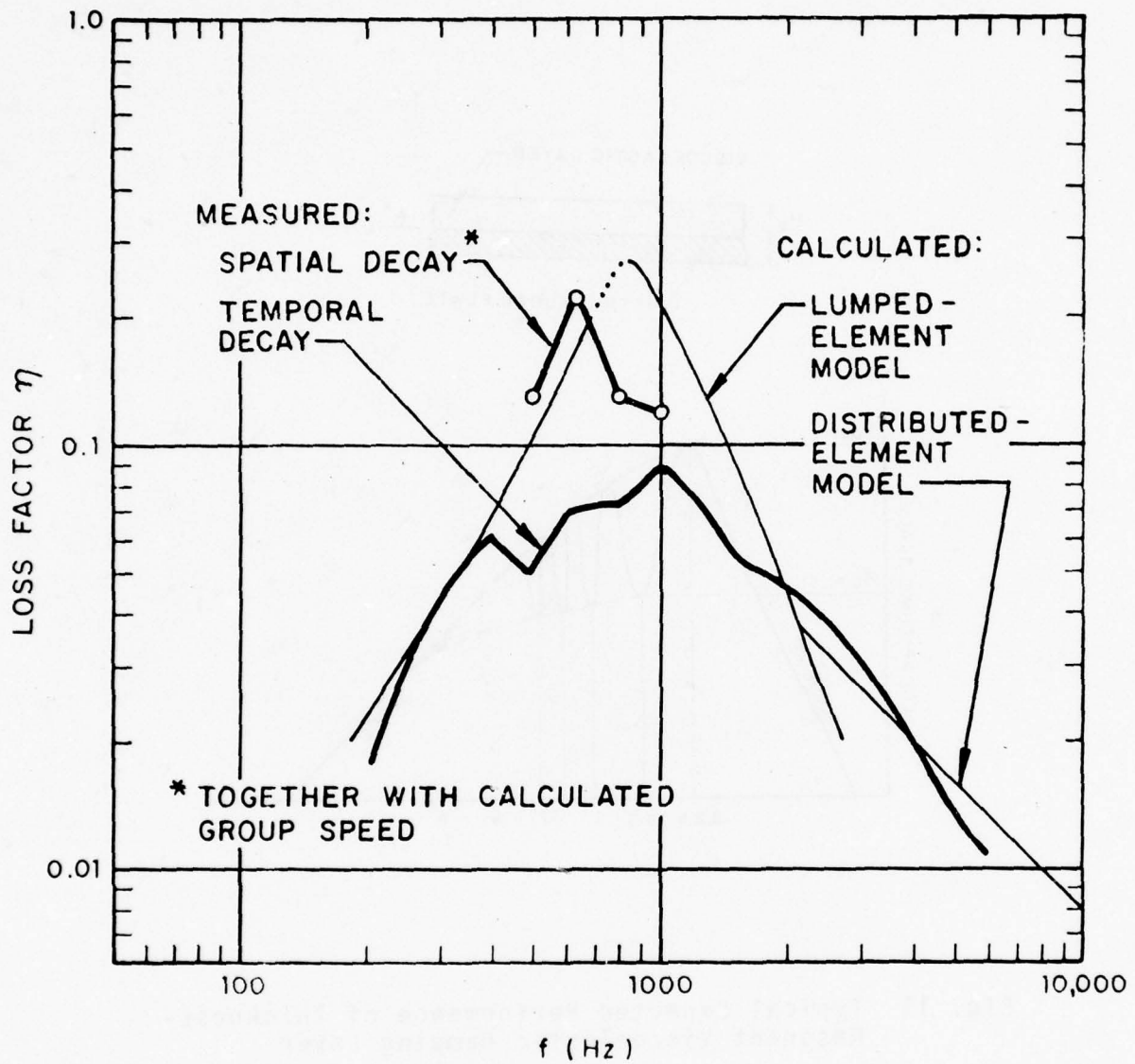


Fig. 12 Measured and Calculated Loss Factor for Water Loaded 1/16-inch Foamed Neoprene Layer on 1/4-inch Steel Plate

FUNDAMENTALS OF SOUND ATTENUATION
IN HOMOGENEOUS POLYMERS AND SIMPLE BLENDS

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"...To a great extent the theory of Sound, as commonly understood, covers the same ground as the theory of Vibrations in general; but, unless some limitation were admitted, the consideration of such subjects as the Tides, not to speak of Optics, would have to be included...."

Lord Rayleigh

The purpose of my lecture today is to relate the basic acoustic parameters, the phase velocity, impedance, and attenuation, to the frequency-dependent mechanical moduli and compliances of polymeric materials. By focusing on the frequency, temperature and pressure dependence of these polymeric mechanical compliances and moduli, we will be able to resolve specific molecular mechanisms for sound attenuation. Time, unfortunately, will not permit us to dwell on the microscopic, molecular details of these processes. My review is primarily directed to the novice student of these matters in the audience. Thus, both limitations in time as well as sound pedagogy force me to select a few basic concepts for your consideration and force me to omit many other relevant topics. We begin with the simplest polymeric materials, the amorphous polymers, and consider in turn the more complicated crystalline polymers and the more or less inhomogeneous polymer blends and related systems.

My talk will cover experimental acoustic measurements over rather different ranges of frequencies in order to bring out fully various mechanisms of sound attenuation. For example, steady wave propagation measurements utilizing travelling wave techniques such as pulse transmission or pulse reflection studies are carried out at fractions to tens of MHz. Dynamic compressibility measurements involve excitation frequencies which are sufficiently low (10^3 - 10^1 Hz) so that the acoustic stress is essentially hydrostatic over the specimen. In these measurements, the specimen is much smaller than the acoustic wavelength. Finally, mechanical measurements involving instruments such as the torsional pendulum bring us into frequencies, which are fractions of a Hz. References 1 and 2 give an excellent

account of the principles and methods underlying these experimental measurements which we cannot review here.

For sufficiently small strains under essentially isothermal conditions, amorphous polymers approximate in mechanical behavior a linear viscoelastic medium (2,3,4). One can propagate in such an extended medium both longitudinal and transverse acoustic waves. In the case of longitudinal waves, the "wave modulus" M^* can, for example, be deduced from pulse transmission measurements, and is given by (4)

$$M^* = M^*(\omega) = M'(\omega) + iM''(\omega)$$

$$\rho c^2 = \rho c^2 / (1 - i\alpha c/\omega)^2, \quad (1)$$

with ρ the density, ω the frequency, c the phase velocity, α the attenuation and $i = \sqrt{-1}$. The real part, or storage modulus, M' ,

$$M' = \rho c^2 (1 - \alpha^2 c^2 / \omega^2) / (1 + \alpha^2 c^2 / \omega^2)^2 \quad (2)$$

is in phase with the oscillating acoustic strain while the imaginary part, or loss modulus M'' ,

$$M'' = 2 \rho c^2 (\alpha c / \omega) / (1 + \alpha^2 c^2 / \omega^2)^2 \quad (3)$$

is 90° out of phase with the acoustic strain. The wave modulus which is related through (1)-(3) to the acoustic parameters (e.g., the acoustic impedance ρc) can be also related to the fundamental mechanical moduli through

$$M^*(\omega) = K^*(\omega) + 4/3 G^*(\omega) \quad (4)$$

where $K^*(\omega) = K'(\omega) + i K''(\omega)$ is the complex, frequency dependent bulk modulus and $G^*(\omega) = G'(\omega) + i G''(\omega)$ is the complex, frequency dependent shear modulus of the viscoelastic medium. The bulk modulus $K^*(\omega)$ is the ratio of the oscillating pressure to the oscillating compressive strain while $G^*(\omega)$ is the ratio of the oscillating shearing stress to the oscillating shear. In either case the in phase, real part of the modulus is the storage modulus and the out of phase, imaginary component is called the loss modulus.

To obtain separately $K^*(\omega)$ and $G^*(\omega)$ one must simply measure the velocity and attenuation of both longitudinal and transverse waves. The wave modulus of transverse acoustic waves, which can be obtained from pulse reflection measurements is $G^*(\omega)$. It should be noted that these expressions for the wave moduli of a viscoelastic medium resemble those of an ideal, isotropic elastic medium with the replacement of the constant elastic modulus K or G by the corresponding frequency dependent complex modulus $K^*(\omega)$

or $G^*(\omega)$, respectively. This is a consequence of the Boltzmann superposition principle of linear viscoelasticity which asserts that the response of the material to a load is independent of the response of the material to any load which is already on the material. An equivalent way of representing the mechanical response follows from this principle: The frequency dependent compliances are simply the reciprocals of the frequency dependent moduli and are the ratio of the complex oscillating strain resulting from the application of a complex oscillating simple stress component, viz.

$$J^*(\omega) = 1/G^*(\omega) = J'(\omega) - iJ''(\omega),$$

$$B^*(\omega) = 1/K^*(\omega) = B'(\omega) - iB''(\omega). \quad (5)$$

As a good approximation the storage compliances J' , B' will appear graphically as the reciprocals of the corresponding storage moduli G' , K' , but because of the minus sign in the definitions given by equation (5), the peaks in the loss moduli G'' , K'' will appear as peaks (and not valleys) in the corresponding loss compliances J'' , B'' . Finally, the remaining two relations of an ideal, isotropic body relating the constant K and G to the constant Young's modulus E and Poisson's ratio ν ,

$$E = 9KG/(3K+G); \nu = (3K-2G)/(6K+2G) \quad (6)$$

are retained for a linear viscoelastic material except that they hold for the complex, frequency dependent moduli.

We now turn to the actual measured values of these moduli and compliances of amorphous polymers (4): In Figure 1 we see dynamic compressibility measurements of the storage bulk compliance B' of natural rubber crosslinked with 12% sulfur as a function of temperature at frequencies from 50-1000 Hz at atmospheric pressure. This plot exhibits the following noteworthy features: (i) At each frequency, the B' vs. temperature graph exhibits an S-shaped curve approaching at high temperatures the typical high rubbery value of the compliances while at sufficiently low temperatures B' achieves a typically smaller value (by about a factor of 1/2) of a glassy polymer. (ii) The apparent transition temperature (at the inflection point) increases with frequency, ω . In principle this transition temperature would extrapolate to the measured glass transition temperature T_{g0} (at one atmosphere) as in the limit ω approaches zero. (iii) All curves have a similar S shape. The region in which the curve is temperature dependent is also the region in which these curves are frequency dependent and vice versa. A positive horizontal shift parallel to the temperature axis would bring curves of lower frequency in coincidence with curves of higher frequency.

Figure 2 shows a plot of both the bulk storage and loss compliance of the same sample of crosslinked natural rubber (5). We note that the loss compliances B'' as a function of temperature possess a single maximum at the same temperature as the inflection point in the storage compliance B' . All this suggests (4,5) that the mechanism of acoustic attenuation is associated with the micro-Brownian motion of the polymeric segments of the backbone of the polymer, whose motion is essentially unfrozen at the frequency ω_r at the temperature of the maximum in B'' or the inflection point temperature of B' . Presumably at atmospheric pressure (excess pressure $P=0$ atmospheres) and as ω vanishes, that temperature ought to be T_{g0} . Figures 3 and 4 show similar behavior of the bulk storage and loss compliances of a sample of poly(vinyl acetate) as a function of temperature at several frequencies and static pressures (6).

The behavior of these compliances suggests strongly that an analog of the "time-temperature equivalence superposition principle" encountered in amorphous polymers is valid here (2,4). Since the frequency dependent moduli and compliances are essentially the Fourier transforms of the time dependent moduli and compliances of linear viscoelasticity, one can more properly denote this a "frequency, temperature, pressure equivalence superposition" principle. It can be stated mathematically for any compliance or modulus of an amorphous polymer by writing

$$B^*(\omega; T, P) = B^*(\omega a_T(T, P)), \dots, \text{etc.},$$

$$G^*(\omega; T, P) = G^*(\omega a_T(T, P)); \quad (7)$$

where $a_T(T, P)$ is the so-called WLF shift factor, a function of the temperature T and excess pressure P by which these curves have to be horizontally shifted along the ω axis to bring them into coincidence (2,4). The form of the shift function follows from a simple "free volume" consideration as:

$$a_T(T, P) = \exp [1/f(T, P) - 1/f_g],$$

$$f(T, P) = f_g + \alpha_f(T - T_{g0}) - \beta_f P, \quad (8)$$

where f is the free volume fraction of the pure polymer, f_g its value at T_{g0} and $P=0$, α_f is the difference in the thermal expansion coefficient of the rubber and the glass and β_f is the corresponding difference in the compressibilities. If the reference temperature is T_{g0} and

the reference excess pressure is $P=0$, then for most polymers one finds that

$$\log a_T(T,0) = \frac{17.44 (T-T_{go})}{51.6 + T-T_{go}} \quad (9)$$

Equation 7 asserts that the complex moduli and compliances are a universal function of the single variable $\omega a_T(T,P)$. Thus, by appropriate scaling of the frequency axis each of these functions can be plotted as a single "master" curve.

It is sometimes easier to refer temperatures to a different reference temperature T_0 , and measure the frequencies from a reference frequency ω_0 . It follows (4) from (7) and (8) that master curves can then be plotted as functions of the reduced frequency ω_r , temperature T_r , and (excess) pressure P_r where:

$$\begin{aligned} \ln \omega_r &= \ln \omega - \frac{\alpha_f (T-T_0) - \beta_f P}{[f_g + \alpha_f (T-T_{go}) - \beta_f P] [f_g + \alpha_f (T_0-T_{go})]}, \\ T_r &= T - \frac{\beta_f P}{\alpha_f} + \frac{1}{\alpha_f} \frac{[f_g + \alpha_f (T-T_{go}) - \beta_f P]^2 \ln(\omega_0/\omega)}{1 - [f_g + \alpha_f (T-T_{go}) - \beta_f P] \ln(\omega_0/\omega)}, \\ P_r &= P - \frac{\alpha_f}{\beta_f} (T-T_0) = \frac{1}{\beta_f} \frac{[f_g + \alpha_f (T-T_{go}) - \beta_f P]^2 \ln(\omega_0/\omega)}{1 - [f_g + \alpha_f (T-T_{go}) - \beta_f P] \ln(\omega_0/\omega)}. \end{aligned} \quad (10)$$

These ideas are tested in the next few figures. Figure 5 shows a plot of the data for the crosslinked natural rubber sample (4) from Figures 1 and 2 as a function of the single variable, the reduced frequency ω_r given by (10). The master curves exhibited in Figure 5 show clearly that the maximum in the master curve for the bulk loss compliance occurs at the inflection point of the bulk storage compliance. Figure 6 is a similar plot for the data on poly(vinyl acetate) (5). Equivalent ways of presenting the same information reflecting the dependence on the reduced temperature T_r and reduced (excess) pressure P_r are shown in Figures 7 and 8 respectively.

Figure 9 shows that the bulk and shear storage moduli of polyisobutylene at 25°C also fall on a master curve when plotted versus the logarithm of the reduced frequency. Figure 10 shows a similar plot for the loss moduli in bulk

and shear of polyisobutylene at 25°C. These data were obtained from pulse propagation and reflection measurements at frequencies from 0.7 to 7.0 MHz (7). Many other polymers investigated, e.g., poly(vinyl chloride), polytrifluoromonoethylen (KEL-F), etc. exhibit responses similar to the ones discussed above. The maxima in the bulk or shear loss moduli occur at the same temperature as the equal inflection points of the storage moduli. Figure 11 illustrates this by a plot of the storage and loss moduli versus temperature for poly(vinyl chloride) (8). This follows as a direct result of the fact that essentially only one molecular loss mechanism need be invoked to explain the attenuation of sound in these materials. Since there are no bulky side groups it is only the motions of the chain backbone which are responsible for the observed attenuation.

Incidentally, the frequency-temperature-pressure superposition principle explains why the moduli and compliances of polymers of sufficiently high degree of polymerization are relatively insensitive to changes in molecular weight. Such a change could affect (7), via (8), only through changes in T_{go} , but T_{go} changes with molecular weight M , only very slowly, roughly like:

$$T_{go} \approx T_{go}^{(\infty)} - \frac{\text{constant}}{M}, \quad (11)$$

where the constant is sufficiently small. Finally the addition of small amounts of organic plasticizers of volume fraction ϕ ought to change the compliances and moduli only through their effect on $\hat{a}_T(T, P, \phi)$ which replaces $a_T(T, P)$ in (7), providing the system remains a single phase. This shift factor a ought to be computed from

$$\hat{a}_f(T, P, \phi) = \exp \left[\frac{1}{\hat{f}(T, P, \phi)} - \frac{1}{f_g} \right] \quad (12)$$

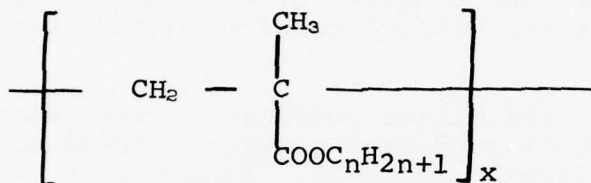
where:

$$\hat{f}(T, P, \phi) = f(T, P) + \gamma \phi,$$

with γ the difference in free volume fraction due to the presence of plasticizer (9).

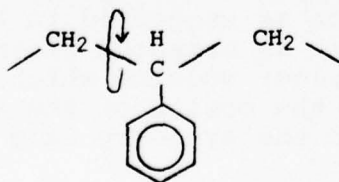
So far the only molecular mechanism for sound attenuation involves the backbone motion of the polymer chain. When a polymer has large pendant groups, the unfreezing of the molecular motions involving the large side group ought to provide a new avenue for dissipating acoustic energy into heat. The extent to which this attenuation

occurs in the propagation of longitudinal sound can be different from what occurs with transverse waves. As a result we expect to see the possibility that the storage moduli exhibit more than one inflection point and that the maxima in the loss moduli in bulk and in shear occur at a different temperature. This is indeed the case, as is shown by the pulse transmission data at frequencies of 0.5-2.25 MHz in several methacrylates of the form:



with increasing number of carbon atoms n in the side chain (9). Figure 12 shows the storage and loss moduli versus temperature for poly(isobutyl methacrylate) which exhibits rather markedly these phenomena. Figure 13 shows how the disparity in the temperature maxima of the loss modulus in bulk and shear varies with number of side chain carbons in the methacrylate series (9). Clearly, secondary dispersion regions may affect differently macroscopic bulk and shear deformations. These secondary dispersions are difficult to disentangle from the primary dispersions due to the micro-Brownian backbone motions at high frequencies. At lower frequencies one can readily distinguish a high temperature, α peak, associated with the T_{g0} , and a lower, β peak, associated with the side chain motions in the loss modulus in shear which is reflected in separate inflection points in the storage modulus. Figure 14 shows the storage and loss moduli for poly(methyl methacrylate) at ~ 0.5 Hz (10).

The low frequency (0.5 Hz) loss spectrum in polystyrene has, besides the α peak (associated with the glass transition), three other peaks. The β peak, which lies below the α peak in temperature, may be due to phenyl group rotation around the main chain, twisting of the main chain or perhaps a crankshaft rotation, illustrated below:



The γ peak which lies below the β peak has been suggested as being due to head-to-head polymerization instead of head-to-tail, i.e., to a chain architectural impurity. Finally the δ peak which occurs at about 70°K is attributed to a rotation of the phenyl group about its linkage to the main

chain, rather than, as proposed for the β , the rotation of the main chain itself (11).

Random copolymers behave generally as one phase amorphous polymer systems with a single glass transition temperature. As a result their moduli resemble that of amorphous homopolymers. Figure 15 shows the moduli of styrene-butadiene rubber versus reduced frequency at a reference temperature of 20°C. These data were obtained at MHz frequencies (12). The behavior of block and graft copolymers is complex and will be considered together with polymer blends.

The behavior of crystalline polymers is very much more complicated than that of amorphous polymers; these are no longer linear viscoelastic materials. Significant departures from the Boltzmann superposition principle appear, particularly at low frequencies. The frequency-temperature equivalence superposition principle also does not apply without extensive modifications. New mechanisms for sound attenuation are now available associated with the crystallites suspended in the amorphous polymer chain matrix. For example, Figure 16 shows the shear moduli of high- and low-density polyethylene as a function of temperature at 1 Hz (13). In low-density (LD) polyethylene there are three loss peaks, two of which can also be found in high-density (HD) polyethylene. The high-temperature α or α' peaks in Figure 16 are associated with crystalline processes since they decrease in intensity when crystallinity is reduced, whereas the low-temperature γ peak increases. The β relaxation is associated with branching and is thus missing in HD polyethylene with its strongly reduced degree of branching. A fair amount of evidence suggests that the γ peak is due to a crankshaft type of rotation involving five or more CH_2 units in the amorphous region between the lamellar crystals (11).

Studies of the velocity of sound transmissions are a very effective way of measuring the amorphous orientation function in crystalline polymers (14). In general, polymer chain orientation (in amorphous or crystalline polymers) affects sound transmission as suggested in Figure 17. The sonic (storage) modulus in an oriented fiber of a polymer is essentially Young's (storage) modulus which varies with the average of the square of the cosine of the angle θ between the sound propagation and the symmetry axis of the units in Figure 17 as shown below:

$$\frac{1}{E} = \frac{1}{\rho c^2} \approx \frac{1 - \overline{\cos^2 \theta}}{E_T^0} \quad (13)$$

with E_0 the intrinsic lateral (transverse) modulus of the perfectly oriented fiber. For the original references to the derivation and validity of this relation, one should consult reference 14. The excellent correlation between the orientation determined from optical birefringence and sonic modulus studies of isotactic polypropylene films, for example, is shown in Figure 18.

Except for rather rare instances where there is a close match of a variety of physical parameters including the solubility parameters, mixtures of two polymers are insoluble in one another and form two-phase systems. Similarly, block and graft copolymers with sufficiently long sequences of each homopolymer are also two-phase systems. In two-phase systems there are two glass transitions instead of the usual one (3). Separating these glass transitions (essentially the same as each homopolymer) is a plateau region whose storage modulus value depends on a) the ratio of the components, and b) which phase tends to be the continuous phase and which is the dispersed phase. Acoustic attenuation in such materials can be quite complex particularly if one or more of polymers in the blend are crystalline. A variety of simplified mechanical models (e.g., Takayanagi) and approximate self-consistent mechanical field equations (e.g., Kerner) have been proposed to deal with such two-phase systems (3,11). We do not have time to review these during this talk. Very useful are exact upper and lower bounds which can be found which apply to two-phase composites: The simplest is if c is the volume fraction of phase 1 with moduli K_1, G_1 ,

$$\begin{aligned} [c/K_1 + (1-c)/K_2]^{-1} &\leq K \leq cK_1 + (1-c)K_2, \\ [c/G_1 + (1-c)/G_2]^{-1} &\leq G \leq cG_1 + (1-c)G_2. \end{aligned} \quad (14)$$

Considerably improved bounds over the primitive estimates shown above have been obtained by Hashin and Shtrikman (15) which apply to the complex moduli of a viscoelastic composite medium. Arridge (11) also reviews the application of these inequalities to polymer blends and block copolymers.

Finally, we mention very briefly a very special class of polymers, the interpenetrating polymer networks (IPN), which have an unusually broad range of both temperatures and frequencies for effective sound attenuation (16). Interesting examples consist of two or more cross-linked chemical networks which are prevented partially or wholly from phase separating by permanent entanglement of the chemically distinct networks. These materials under certain circumstances exhibit only one very broad glass transition. Latex IPN's already were prepared in 1969 (17), but more recently a number of new latex IPN's have been synthesized

to produce what is termed "silent paints", two layer coating systems capable of attenuating noise and vibration over a broad temperature range (18). Certain filled IPN foams (19) appear to be even more promising.

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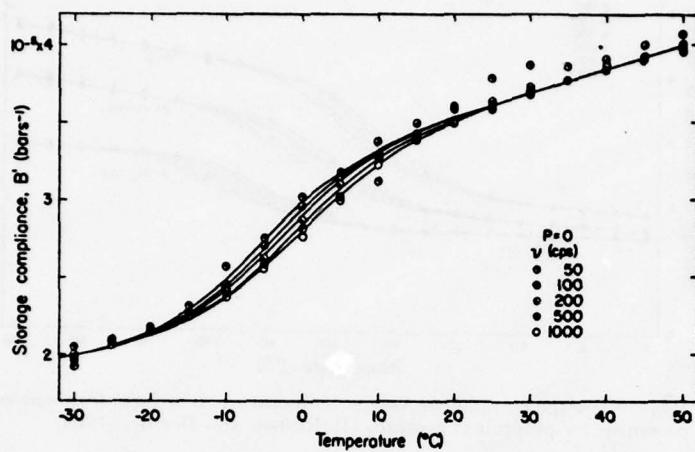


Fig. 1. Storage compliance versus temperature at atmospheric pressure and several frequencies for natural rubber-12% sulfur (McKinney *et al.*, 1960).

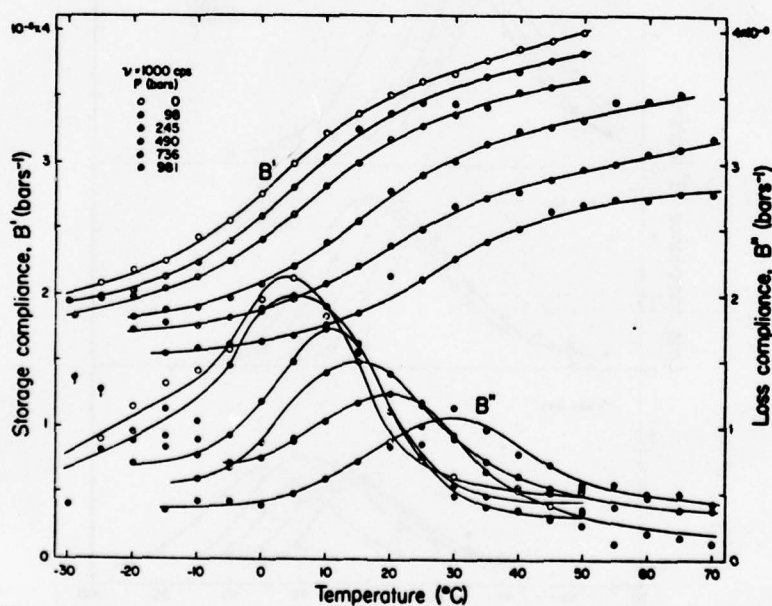


Fig. 2. Storage and loss compliances versus temperature at 1000 cps and several static pressures for natural rubber-12% sulfur (McKinney *et al.*, 1960).

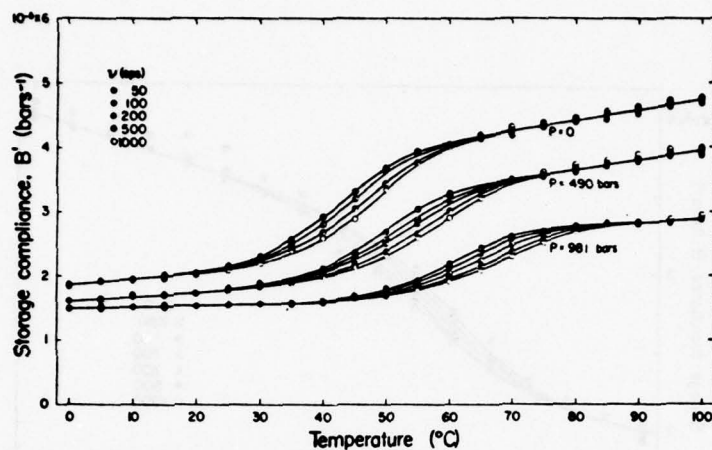


Fig. 3. Storage compliance versus temperature at several frequencies and static pressures for poly(vinyl acetate) (McKinney and Belcher, 1963).

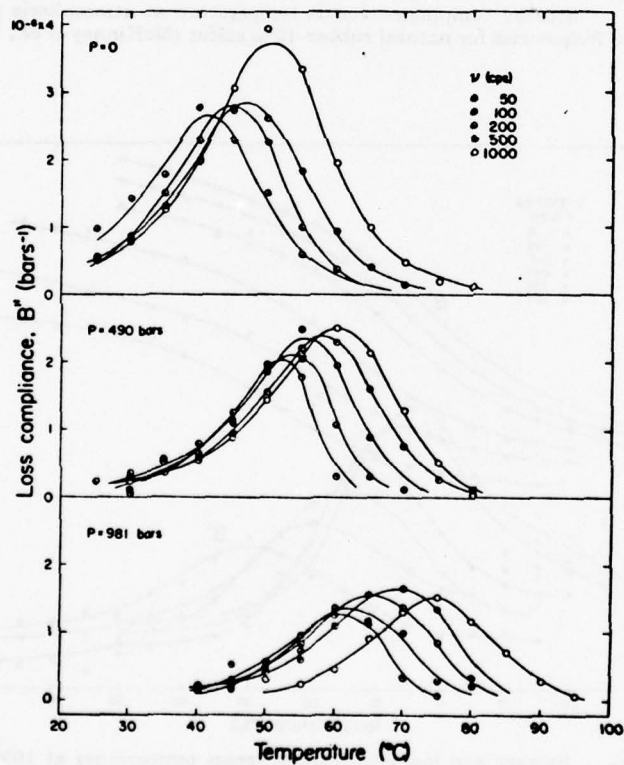


Fig. 4. Loss compliance versus temperature at several frequencies and static pressures for poly(vinyl acetate) (McKinney and Belcher, 1963).

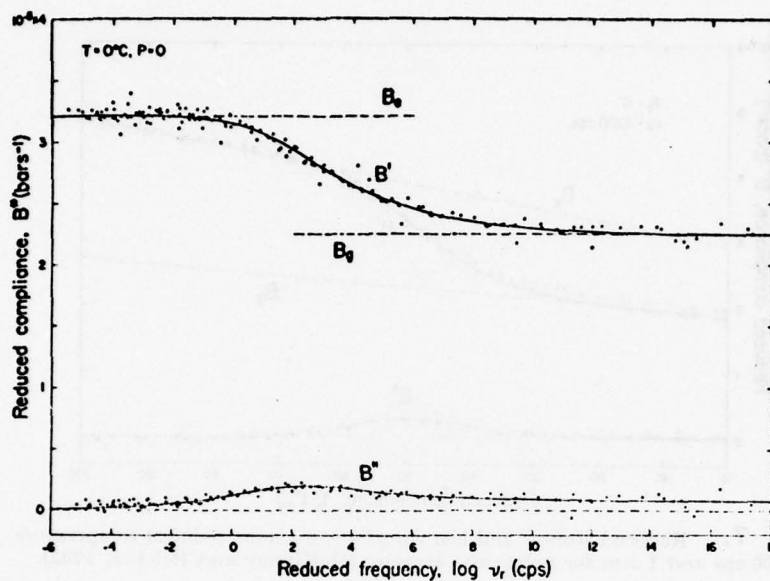


Fig. 5. Reduced storage and loss compliances versus frequency at 0°C and 1 atm for natural rubber-12% sulfur (McKinney *et al.*, 1960).

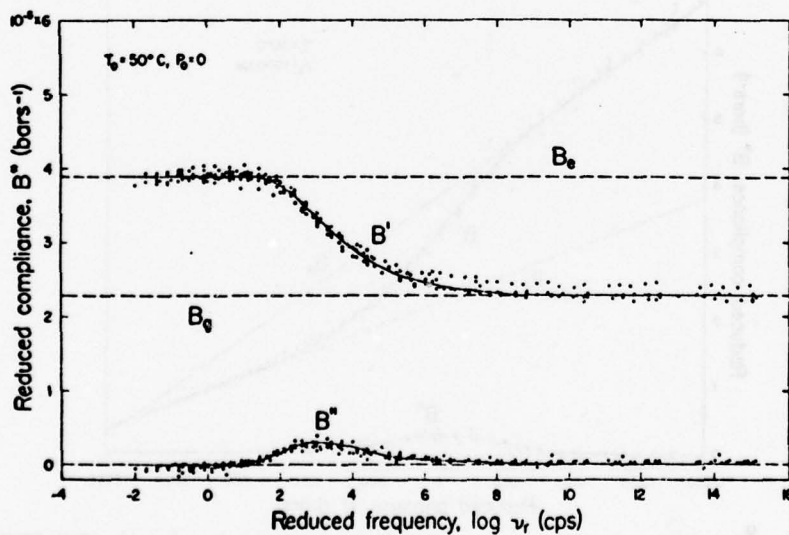


Fig. 6. Reduced storage and loss compliances versus reduced frequency at 50°C and 1 atm for poly(vinyl acetate) (McKinney and Belcher, 1963).

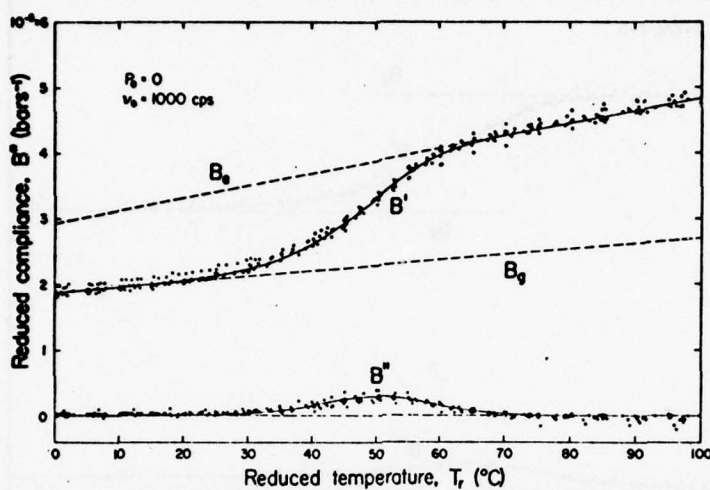


Fig. 7. Reduced storage and loss compliances versus reduced temperature at 1000 cps and 1 atm for poly(vinyl acetate) (McKinney and Belcher, 1963).

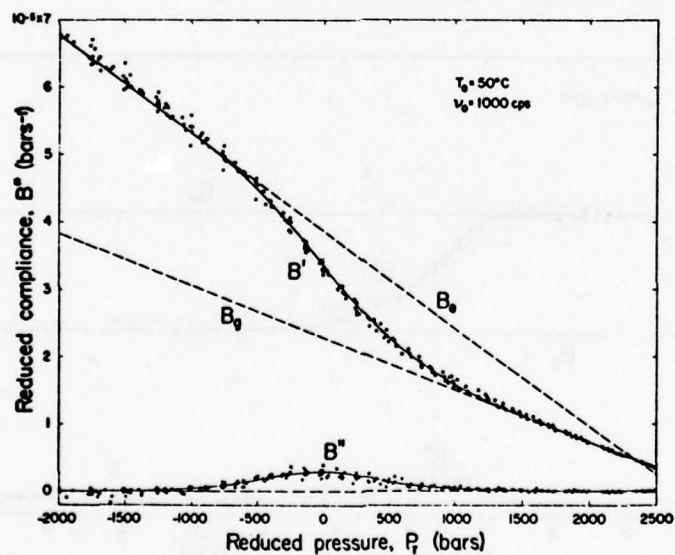


Fig. 8. Reduced storage and loss compliances versus reduced static pressure at 1000 cps and 50°C for poly(vinyl acetate) (McKinney and Belcher, 1963).

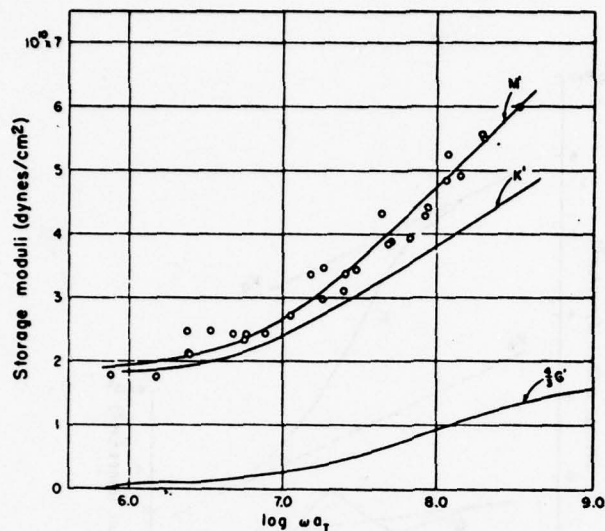


Fig. 9. Reduced storage moduli versus reduced frequency at 25°C for polyisobutylene. K = bulk, G = shear, $M = K + (4/3)G$ (Marvin *et al.*, 1954).

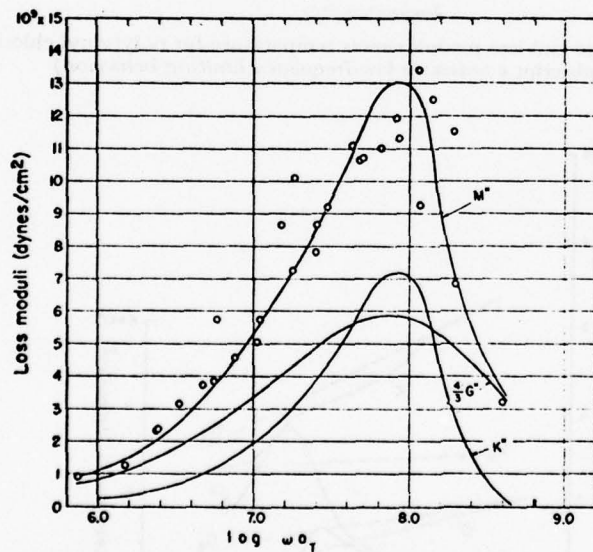


Fig. 10. Reduced loss moduli versus reduced frequency at 25°C for polyisobutylene (Marvin *et al.*, 1954).

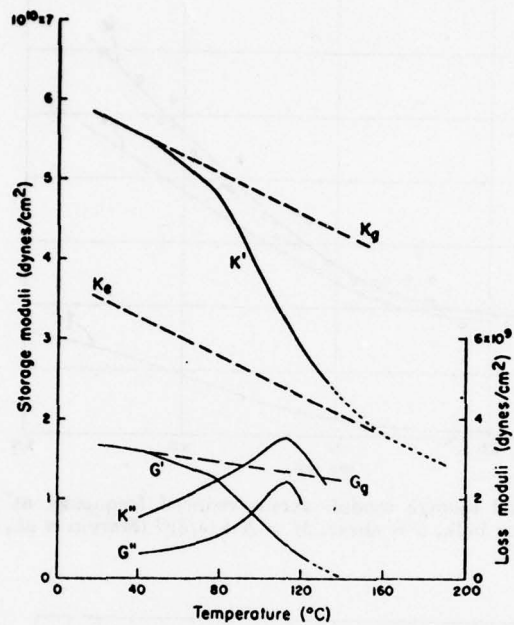


Fig. 11. Storage and loss moduli versus temperature for poly(vinyl chloride) (Kono, 1961a). (Subscript *e* indicates low-frequency limiting behavior.)

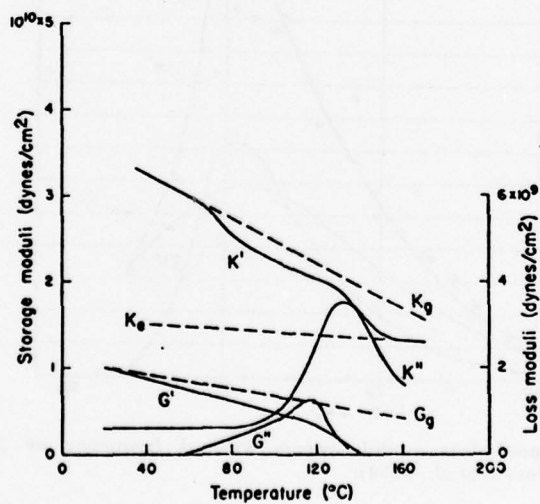


Fig. 12. Storage and loss moduli versus temperature for poly(isobutyl methacrylate) (Kono, 1961a).

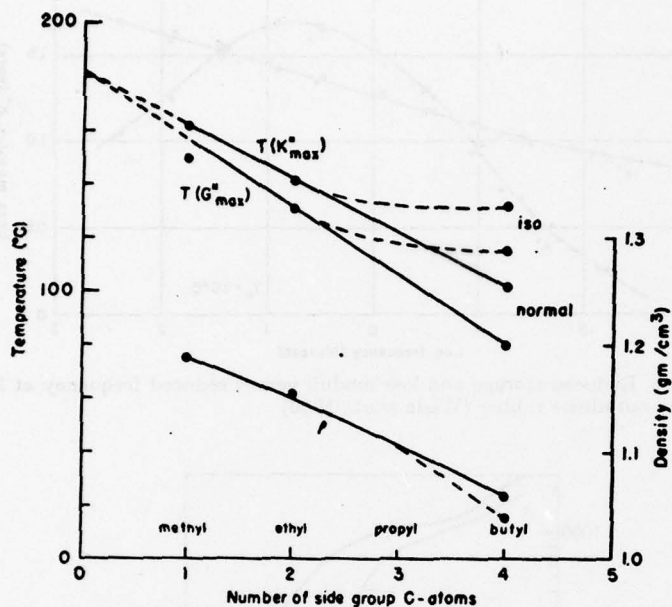


Fig. 13. Density, $T(K'_{\max})$, and $T(G'_{\max})$ versus the number of side group C-atoms for the methacrylate series (Kono, 1961a).

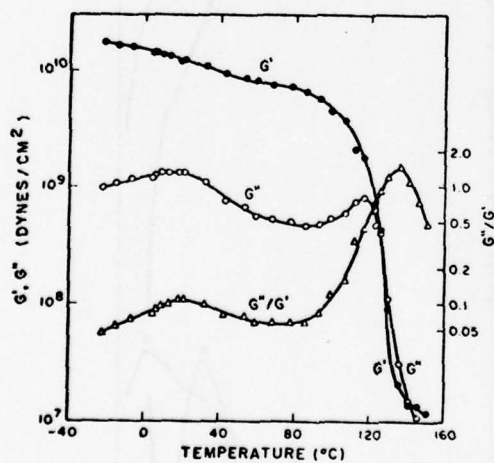


Fig. 14. G' , G'' , and $G''/G' = \tan \delta$ for polymethyl methacrylate at ~ 0.5 cps. (After Iwayanagi and Hideshima, 1953a.)

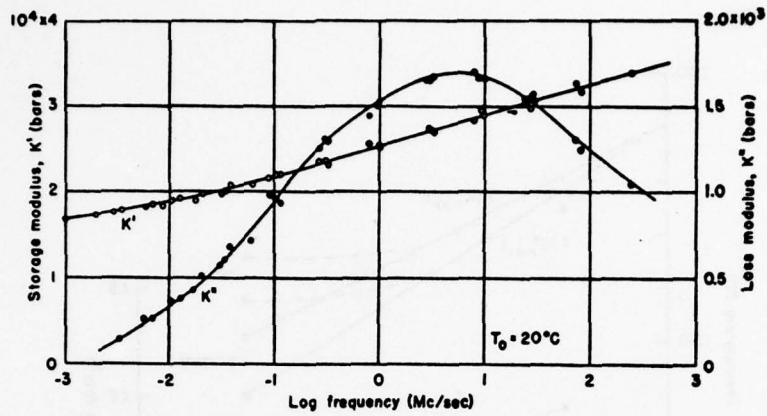


Fig. 15. Reduced storage and loss moduli versus reduced frequency at 20°C for styrene-butadiene rubber (Wada *et al.*, 1960).

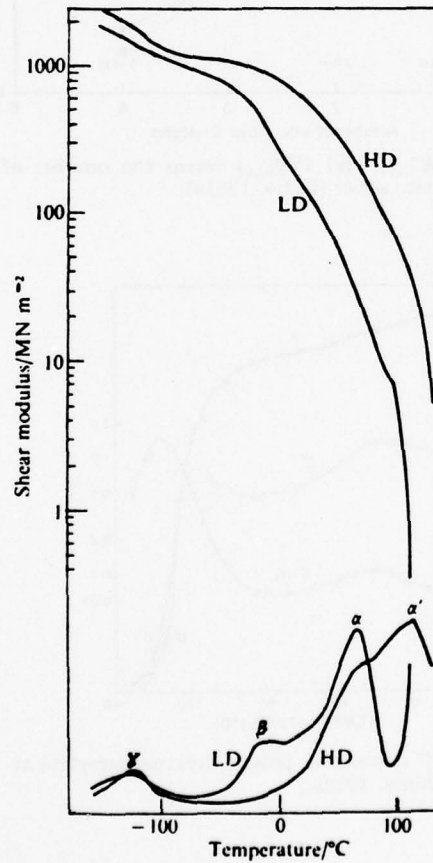


Fig. 16. The modulus and loss behaviour of polyethylene as a function of temperature. (Flocke 1962)

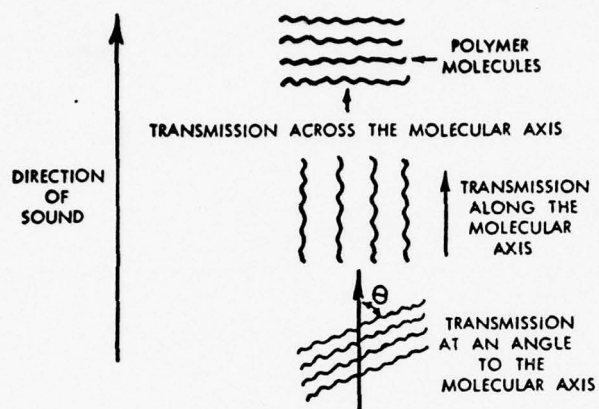


Fig. 17. Possible modes of sound transmission in polymers.

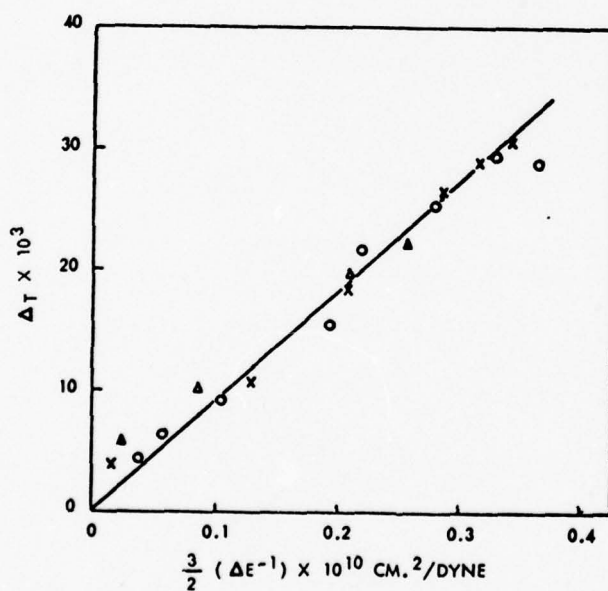


Fig. 18. Relation between the birefringence and the sonic modulus of isotactic polypropylene films: (O) Series A; (x) Series B; (Δ) Series C

DAMPING AND MECHANICAL PROPERTIES OF FILLED
POLYMERIC SYSTEMS

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Unfilled Polymers

The dynamic mechanical properties (the imaginary part of the elastic modulus G'' , the real part of the elastic modulus G' , and their ratio G''/G') of most amorphous, uncrosslinked polymers are similar to the results shown in Figure 1 (1-3). In this case, G' is the shear modulus while G'' and G''/G' are energy dissipation or damping terms. At frequencies of 1 Hz or less, the peaks in G'' and G''/G' occur at temperatures associated with the glass transition temperature T_g of the polymer. The upswing in the G''/G' curve above T_g is due to the increasing importance of viscous behavior relative to the elasticity at the higher temperatures. The energy dissipated into heat for a unit deformation of the polymer is greatest near T_g and at lower temperatures since this heat generated is proportional to G'' . However, free vibrations will damp out most rapidly where G''/G' is greatest, that is, near T_g and at the very highest temperatures shown in Figure 1. The curves shown in Figure 1 are for some constant frequency; these curves will shift to higher temperatures as the frequency of vibration increases. Typical values are about 7°C shift in the G''/G' peak for a factor of ten increase in the frequency. The temperature where the damping peak is a maximum also can be shifted by adding a plasticizer or by forming a copolymer. The glass transition T_g of a polymer can be varied from below -120°C to over 300°C by changing the chemical structure of the polymer.

Polymers Containing Rigid Fillers (Including Syntactic Foams)

Fillers can bring about dramatic changes in the modulus and damping behavior of a polymer (3-7). Figure 2 shows the typical behavior of a filled polymer compared to the unfilled polymer. Rigid fillers increase the elastic modulus, especially near T_g and above. Fillers nearly always increase the damping G''/G' above T_g , but below T_g fillers may either increase or decrease G''/G' . Fillers also may shift the damping peak to higher temperatures in some cases.

Many equations have been developed for predicting the moduli of filled systems. One such set of equations is (3,8-10):

$$\frac{M}{M_1} = \frac{1 + AB\phi_2}{1 - B\psi\phi_2} \quad (1)$$

$$B = \frac{M_2/M_1 - 1}{M_2/M_1 + A} \quad (2)$$

$$\psi \doteq 1 + \left(\frac{1 - \phi_m}{\phi_m^2} \right) \phi_2 \quad (3)$$

The modulus (either shear or Young's) is M ; the subscripts 1 and 2 refer to the continuous phase and the dispersed phase, respectively. The constant A depends upon the shape of the particle, its orientation, and upon the degree of adhesion at the interface. The constant B is defined by equation 2. The reduced concentration factor ψ depends upon the maximum packing fraction ϕ_m of the filler. The volume fraction of filler is ϕ_2 . Tables of values for A and ϕ_m are given in reference 3.

The damping of a filled system when the filler does not give rise to any additional damping mechanisms is approximately (3,11):

$$\frac{M''}{M'} = \left(\frac{M''}{M'} \right)_1 \phi_1 + \left(\frac{M''}{M'} \right)_2 \phi_2 \quad (4)$$

Some scientists believe the damping of such a filled system should be greater (12,13). In most cases, the damping of the filler $(M''/M')_2$ is very small. Certainly, in such cases if $(M''/M')/(M''/M')_1 \phi_1$ is considerably greater than 1.0, the filler must be responsible for the introduction of new damping mechanisms. New damping mechanisms which can be due to the filler are particle-particle motions and polymer-particle slippage. How can one maximize the damping M''/M' by using these mechanisms?

Unless there is motion of particle-particle contact points or slippage at the polymer-particle interface, the

addition of a rigid filler to a polymer generally decreases the damping. The damping can be increased by bringing about particle-particle motion or by increasing the number of such contact points. A way of doing this is to produce weak agglomerates of the filler particles in the polymer. This can be accomplished in several ways. First, the filler may be poorly mixed with the polymer so that the filler is poorly dispersed. Second, the filler surface can be treated with an agent which promotes agglomeration of the filler particles. Special silane coupling agents or surfactants which make the filler particles "like" each other better than they "like" the polymer will simultaneously produce agglomeration and destroy polymer-filler adhesion. To obtain the maximum damping it is important that the stresses induced into the filler by a deformation of the plastic matrix are great enough to overcome the mechanical friction at the points of contact. Both these stresses and the mechanical friction vary with temperature. Below T_g , a deformation of the polymer exerts very high stresses on the filler particles because of the high modulus of the polymer. In this case, the damping below T_g will be high unless the agglomerated particles are strong or unless the friction between particles is very high. Above T_g , the polymer is capable of exerting only small forces on the filler particles, but the damping can still be high under certain conditions. Because of the mismatch in the thermal coefficients of expansion of the polymer and filler, the polymer exerts less "squeezing" force on the polymer as the temperature is raised. Thus, as the temperature is raised above T_g , the particle-particle friction and the polymer-filler friction both decrease, and smaller stresses are needed to produce the relative motion at particle surfaces that is essential for high damping.

The importance of the nature of the particle interface is illustrated in Figure 3 (14). A treatment which promotes good dispersion and strong polymer-filler adhesion (curve A) decreases the damping of glass beads in polystyrene. Treatment of the glass beads with a silane that destroys adhesion and promotes particle agglomeration (curve C) greatly increases the reduced damping. Note that in all three curves the reduced damping is greater than 1.0 so that the filler is inducing some extra damping even in curve A. It is very difficult to separate the effects of particle-particle motion from polymer-particle slippage. However, in Figure 3 it is believed that particle-particle motion plays a bigger role than polymer-particle slippage, but curve C might be an exception. Additional types of data are required in order to make a definite decision, however.

Changes in the temperature can produce some unexpected phenomena, especially in composites in which the filler particles are agglomerated. Figure 4 shows the general

behavior of the relative modulus M/M_1 for filled polymers (15). The theories predict a relative modulus which is essentially independent of temperature except for a small jump near T_g . Sometimes, however, the experimental values of the relative modulus show very large changes with temperature. It is now known that this temperature dependence is due to a combination of several effects (15,16). Below T_g , the temperature dependence of the relative modulus results from the difference in the coefficients of thermal expansion of the two phases (16). Near T_g there is a jump in the relative modulus, and it goes through a pronounced maximum. Most of this jump is due to agglomerated particles or particle-particle contacts which show relative motion when the modulus of the polymer is high but which become rigid at slightly higher temperatures where the modulus of the polymer is much less (15). At still higher temperatures the relative modulus decreases and the reduced damping increases with temperature. These phenomena again are largely due to the mismatch in coefficients of thermal expansion which reduces friction and allows more particle-particle motion as the temperature is increased (15). It should be noted that the temperature dependence of the relative modulus is the opposite of the behavior of the reduced damping. For example, if the relative modulus decreases with an increase in temperature, the reduced damping will increase with temperature.

The jump in relative modulus and the behavior above and below T_g can be represented by modifying equation 1 to:

$$\frac{M}{M_1} = \frac{1 + \bar{A}B\phi_2}{1 - B\psi\phi_2} \quad (5)$$

where:

$$\bar{A} = \frac{A'\phi_2' + A''\phi_2'' + A'''\phi_2''' + \dots}{\phi_2} \quad (6)$$

In equation 6, A' is a constant for the perfectly dispersed particles and ϕ_2' is their concentration. $A' = 1.5$ for spheres with perfect adhesion to the matrix while $A' = 0$ for spheres with no adhesion. A'' is a constant which determines the effect of two particles in contact, and ϕ_2'' is the concentration of such doublets. A''' is the constant for agglomerates made up of three particles, and ϕ_2''' is their concentration. The value of A for large rigid spherical agglomerates can approach 4.0, while the value for weak

agglomerates is very small. For a given filled system, the values of A' , A'' , etc. change with temperature, depending upon whether or not the agglomerates are rigid or "soft" and on whether or not the adhesion at the interfaces is high or low (15). The concentration of agglomerates increases as the total concentration of filler increases.

In summary, it has been shown that the damping of filled systems can be varied over a very wide range of values by controlling the following factors: 1. The state of agglomeration of the filler particles. 2. The resistance of the agglomerates to fracture or to particle-particle motion. 3. The coefficient of friction for one particle moving against another. 4. The degree of adhesion of the polymer to the filler particles.

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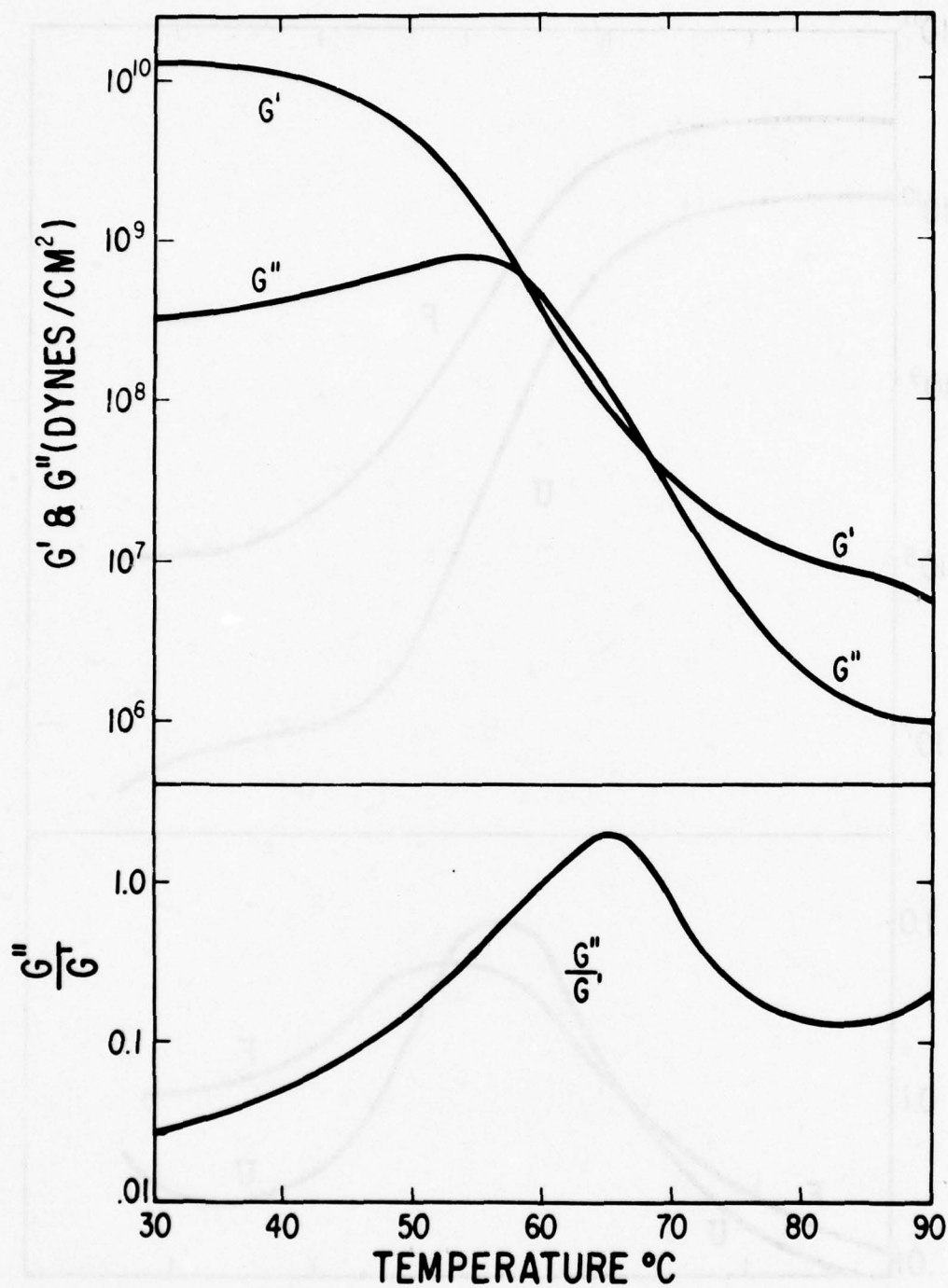


FIGURE 1.

The general behavior of the dynamic mechanical properties of amorphous polymers as a function of temperature. G' is the shear modulus, and G'' is the shear loss modulus or the imaginary part of the shear modulus.

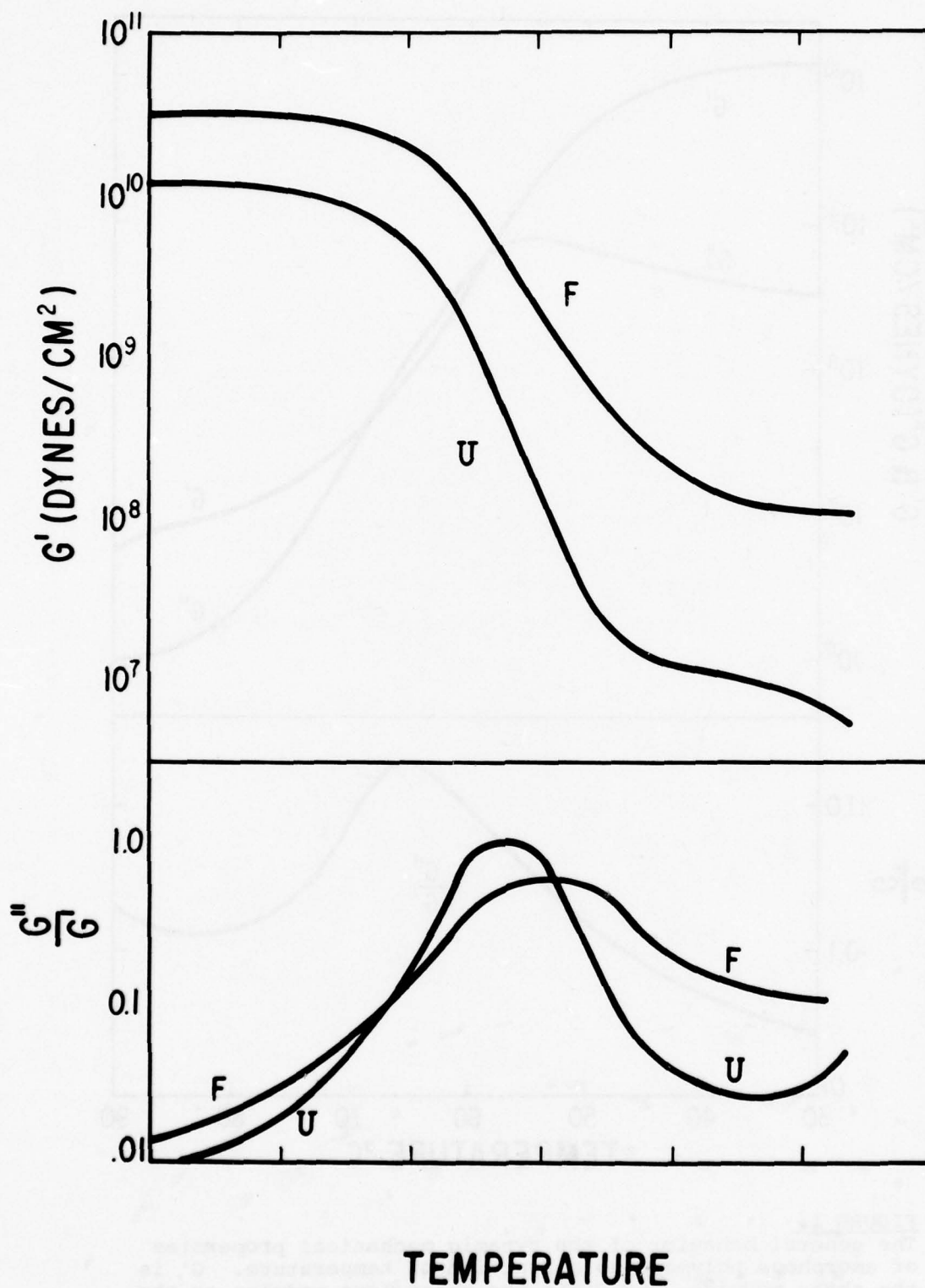


FIGURE 2.
The general dynamic mechanical behavior of a filled polymer F compared to an unfilled polymer U. [From Shock and Vibration Digest, 7, 1 (Feb. 1975).]

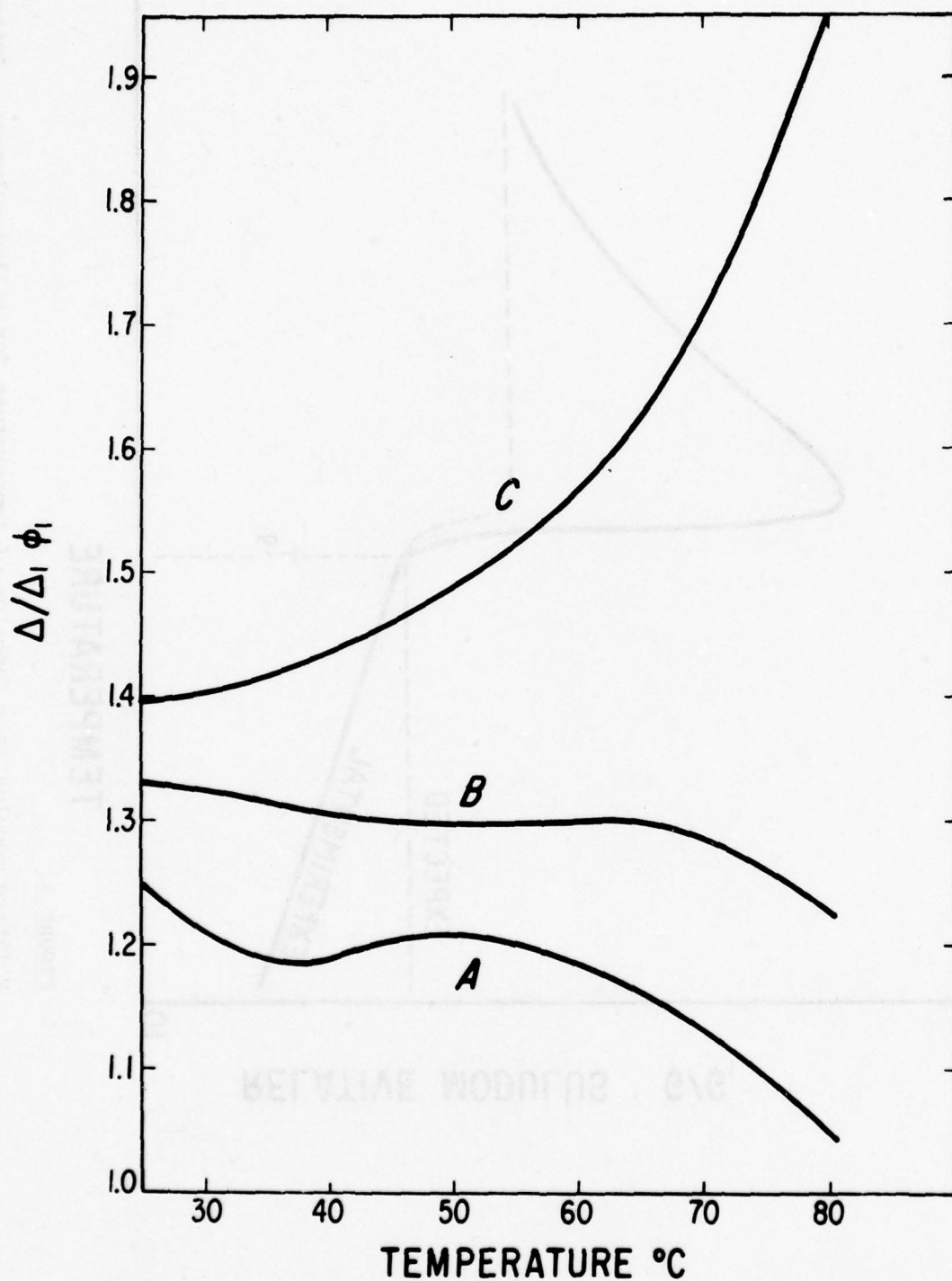


FIGURE 3.
 Reduced damping of polystyrene filled with glass beads to a volume fraction of 35%. A. Beads treated with A-1100 silane to promote adhesion. B. Untreated beads. C. Beads treated with dimethyl dichlorosilane to promote agglomeration and to destroy adhesion. $\Delta = G''/G'$. [Modified from the data of Nielsen and Lee, J. Composite Mater., 6, 136 (1972).]

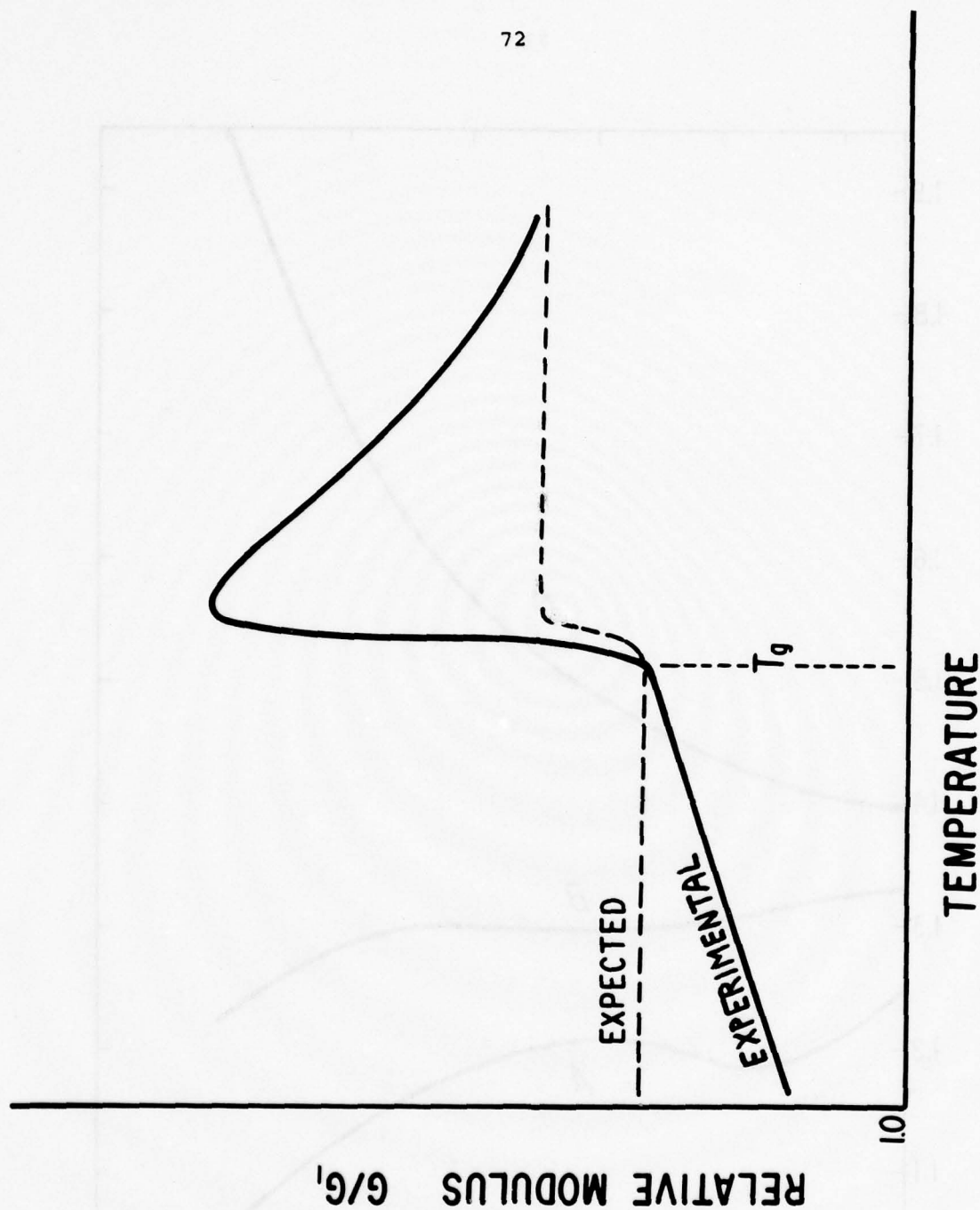


FIGURE 4.

Relative modulus as a function of temperature for filled polymers. Dotted line shows the expected behavior based on the theories of dispersed particles in a matrix. Solid line shows the usual experimental behavior for real systems which contain agglomerated particles and which have mismatched coefficients of thermal expansion. From Lee and Nielsen, J. Polym. Sci. (Phys.), 15, 683 (1977).

MATERIALS AND PROCESS
FOR
STRUCTURAL FOAMS

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Various organic foams have been used for structural applications for many years. The aerospace and marine industry use urethane and syntactic foams; the building industry utilizes urethane, phenolic, and vinyl foams. The automobile industry uses many types of foams primarily for insulation and acoustic attenuation. Recently, the syntactic foams have been receiving attention not only for structural applications but for acoustic attenuation purposes. Table I shows typical strength properties of commonly used structural foams. As can be noted, the syntactic foam is 300% more efficient than the other foams listed for both compression and shear loading. In flexure the syntactic foam is 40% more efficient than the urethane foam.

Typical uses of such foam materials have been as core materials. The syntactic foams are used to hold inserts and seal edges of sandwich panels with honeycomb cores. The syntactic foams have also been used in place of the honeycomb in sandwich panels to stabilize the skins. These foams are also used as lightweight shims or spacers between multiple members that are to be joined by bolting. An important use for syntactic foams is for buoyancy applications such as filler for voids and encasement for instrument packages operating at various ocean depths. Syntactic foams can also be formulated to attenuate acoustic energy and thus are used for acoustic windows and attenuators.

Presently, several ranges of density exist for epoxy syntactic foams. The foams are based on either glass microsphere or macrosphere fillers in an epoxy matrix. Other fillers can be incorporated to modify the mechanical and acoustic properties of the material. For structural applications in the marine field, the material is specified by MIL-S-24154A. The properties of the specification grade materials are given in Table II.

Significantly lower densities of epoxy syntactic foams can be prepared from a careful mixture of various sizes of hollow spheres. To date densities as low as 18 pounds per cubic foot have been obtained, but naturally the strengths

are low. Various types of hollow macrospheres are possible; Figure 1 shows a technique used by the 3M Company.

Various processes are used to prepare or cure syntactic foams for structural applications. The most common method is to mix and pour. The micro/macrospheres are mixed with catalyzed resin and poured by gravity into a mold and heat cured. A second method is called wet pack. Catalyzed resin is used to precoat the micro/macrosphere fillers and packed into a mold under pressure. Additional catalyzed resin is added to close all voids and the material is heat cured. The third method is termed dry pack. In this method, the micro/macrospheres are packed into a mold by vibration, catalyzed resin is infiltrated by vacuum or pressure, and the system is then heat cured.

The cured shapes of syntactic foam can then be machined into the final desired shape. Circular or band saws with high speed steel or carbide blades operating at high cutter speed and slow feed with water coolant provide high tolerance cuts. The material can be accurately milled or routed with carbide-tipped tools. Tool bit speed should be high and workpiece feed should be slow. Single point cutters or grinding tools can be used. Syntactic foams are drilled to close hole tolerances with carbide- or diamond-tipped drills. Moderate drill speed and slow feed are recommended as well as backing out the drill to clear the hole and cooling with water. The syntactic foam can be sanded with either belt or disc sanders using silicon carbide paper of any desired grit size.

Syntactic foams can be joined by adhesive bonding or with mechanical fasteners. Heat-curable thixotropic paste or film adhesives have been successfully used. One prepares the foam for bonding by sanding and remove sanding dust with clean dry lint free wipers. As an alternative, the foam can be prepared for bonding by dry honing. The use of adhesive primers is optional. Syntactic foams can be joined by mechanical fasteners if techniques are used to prevent cracking and crushing. Holes should be of a close tolerance with minimum $3D$ edge distance and $2.5D$ side distance and $5D$ pitch. The joint should be designed for bearing with the bearing allowable at the 4% hole elongation point. The fasteners are installed with washers and the installation torque is controlled. The use of impact or self threading fasteners should not be considered.

Quality control procedures for syntactic foams used for structural applications include material acceptance, in process inspection, and final non-destructive testing. Material acceptance tests include properties such as density, water absorption, and compression strength. In process controls include dimensional, weight, acoustic

impedance, and bond preparation. Final quality tests cover ultrasonic inspection for density uniformity, radiography or ultrasonics for joint integrity, and dye penetrants for edge microcrack freeness.

ADVANCED SYNTACTIC MATERIAL FORMS

Syntactic foams can be used for acoustic attenuation when they are specially formulated for a given acoustic energy level. In an attempt to provide standardized materials, the syntactic foam could be formulated to a B-stage prepreg of various thickness in a manner not unsimilar to glass-fiber reinforced plastics or film adhesives. A typical series of such standardized syntactic prepreps is shown in Figure 2. The prepreg materials could be joined, one over the other, to create a mixed layer to attenuate acoustically a specified energy level. The stacked prepreg could be installed into a mold and under heat and moderate pressure effect lamination to any desired shape. A typical stack-up or custom-design material is shown in Figure 3.

CONCLUSIONS

Syntactic foams are effective materials for various structural applications for many industries. The foams can provide a broad class of materials for many applications; presently, the more important of which are as core materials, buoyancy materials, and more recently, for acoustic attenuation purposes. Special syntactic materials have been developed for ablation applications and high temperature insulation. It is certain that these materials can be further optimized for structural as well as specialty applications.

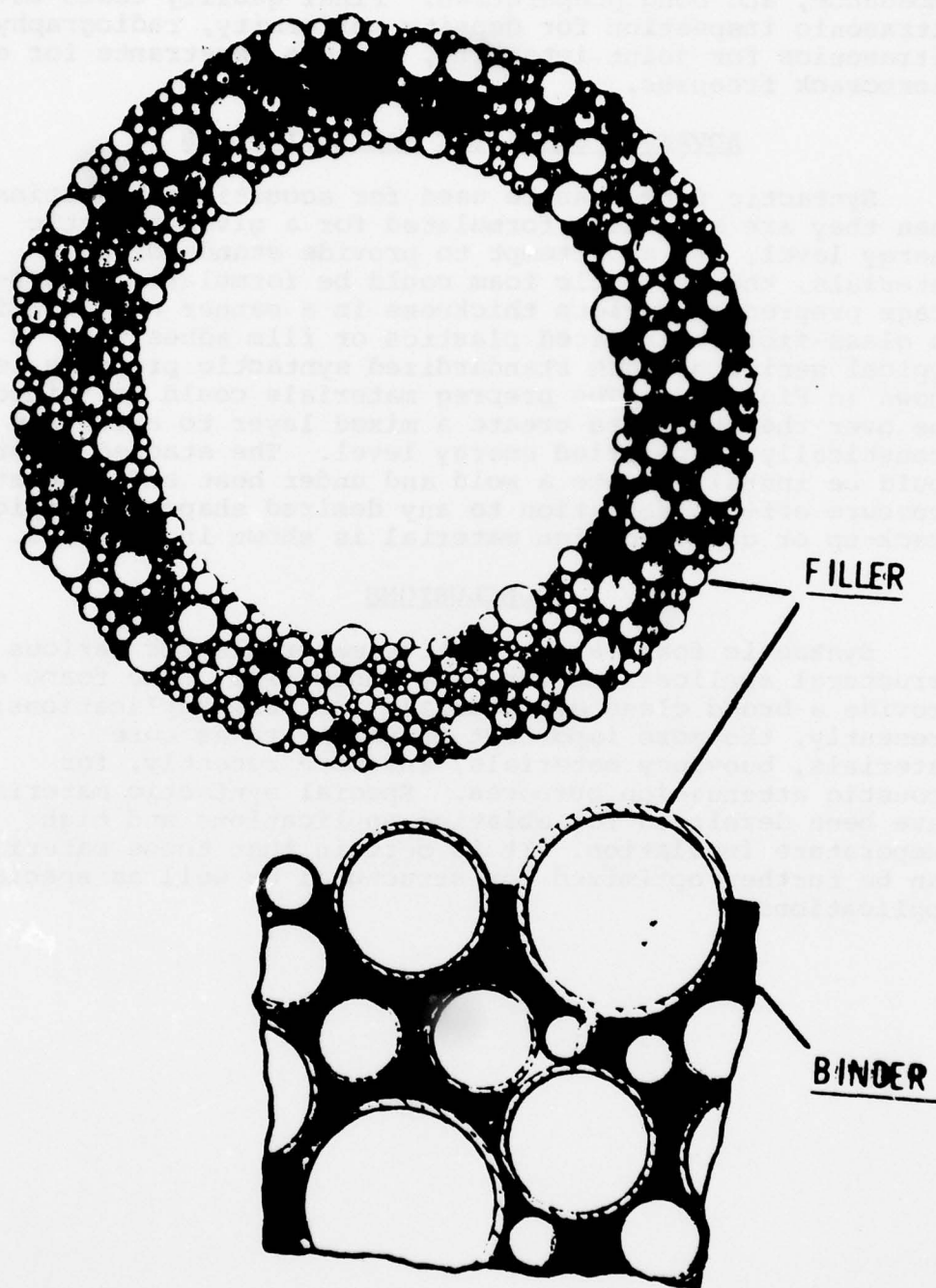


FIGURE 1 STRUCTURE OF "3M" BRAND HOLLOW MACROSPHERE

STANDARDIZED PRODUCTS

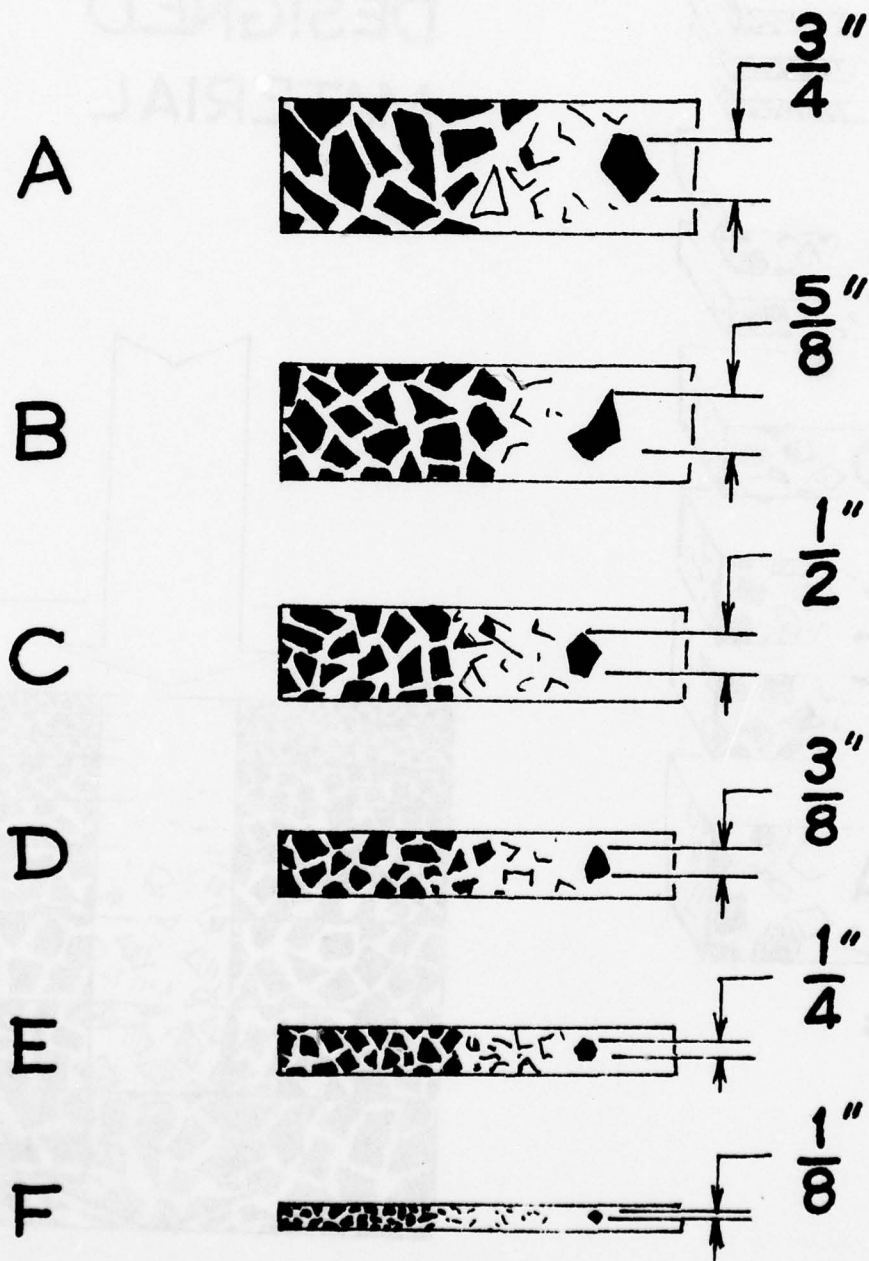


FIGURE 2

CUSTOM DESIGNED MATERIAL

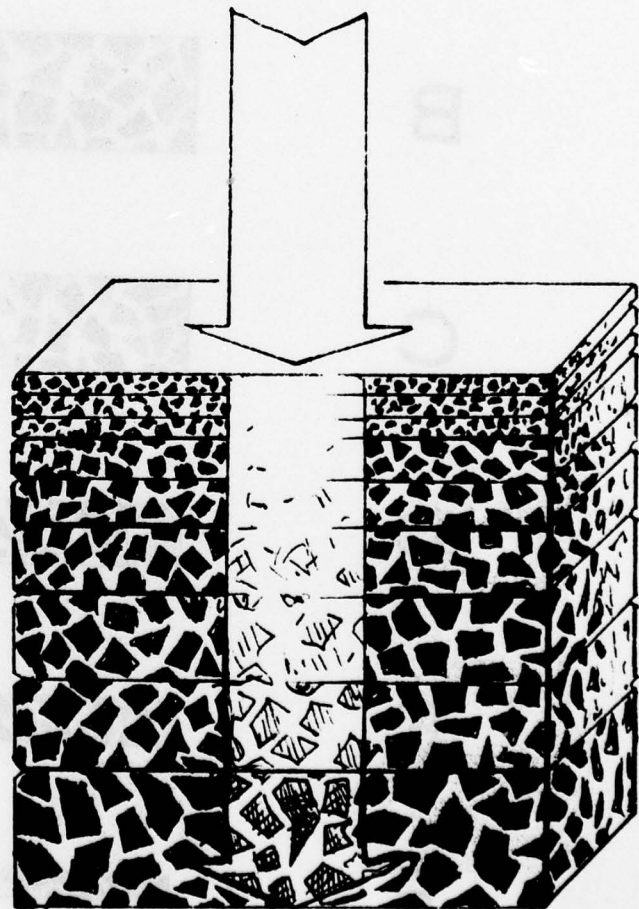
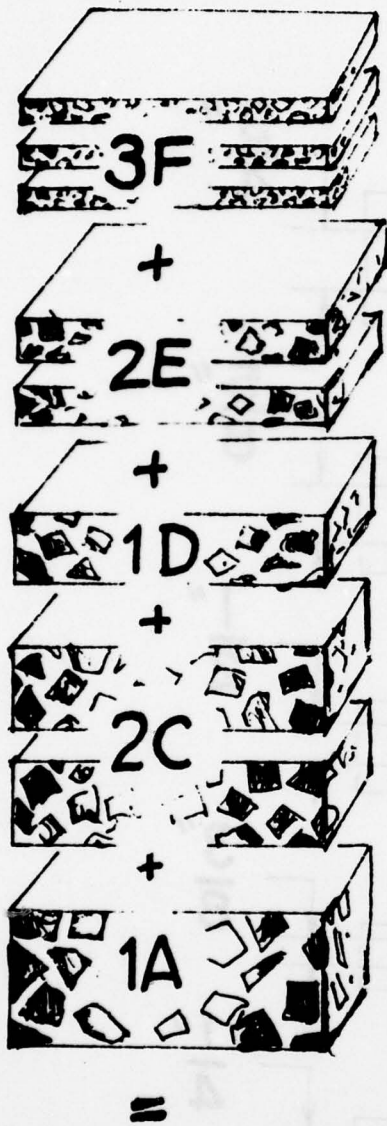


FIGURE 3

Table I

STRENGTH TO WEIGHT RATIOS OF VARIOUS RIGID FOAMS

<u>Property</u>	<u>Epoxy Syntactic</u>	<u>ABS</u>	<u>Epoxy</u>	<u>Phenolic</u>	<u>Vinyl</u>	<u>Urethane</u>
Density, pcf	36	40	20	22	3	24
Compression Strength, psi	9500	2300	1080	1200	80	2000
Compression Strength to Density	264	58	54	55	27	83
Flexural Strength, psi	3500	3000	--	--	--	1600
Flexural Strength to Density	97	75	--	--	--	67
Shear Strength, psi	3500	--	--	--	--	700
Shear Strength to Density	97	--	--	--	--	29

Table II

STRUCTURAL GRADE EPOXY SYNTACTIC FOAM PROPERTIES
AT ROOM TEMPERATURE

	<u>HOLLOW MICROSPHERE FILLED EPOXY</u>	
	<u>Type I</u>	<u>Type II</u>
Density, lb/ft ³	36	42
Uniaxial Compression Str., ksi	9.5	14.5
Compression Modulus, ksi	370	450
Tensile Strength, ksi	3.0	5.0
Shear Strength, ksi	3.5	5.0
Flexural Strength, ksi	3.5	6.5
Bulk Modulus, ksi	325	450
Water Absorption, % by wt.	2.1	1.5

APPLICATIONS TECHNOLOGY SESSION

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INTRODUCTORY REMARKS

The preceding four presentations gave a comprehensive review of the relevant acoustical principles and of the relationships between material composition and structure on the one hand and mechanical properties on the other. The application of these basic concepts to practical systems is quite complex. First, the desired acoustical functions vary widely from one application to another. Commonly, what is needed is merely damping of mechanical vibrations; this is often the case for parts subjected to forced vibrations. In other cases, the requirement is for vibration isolation, or attenuation of transmitted noise. Still in other cases, the requirement is for suppression of sound reflection.

The approaches used in practice vary just as widely. In the case of forced vibrations, the time-tested approaches include elimination or change in frequency of the forcing source; changes in inertia and stiffness in order to change the natural frequency of the vibrating part; the introduction of viscous, viscoelastic, and frictional forces in order to provide viscous damping; and the addition of suitable components capable of providing dynamic damping. Vibration isolation has been achieved by a variety of means, usually combining one or more viscoelastic components (usually rubber) with suitable elastic and inertial features. The variety of approaches available for the attenuation of reflected noise have been covered in the preceding session.

The next two papers will illustrate some of these practical problems in the case of aerospace and naval applications.

APPLICATIONS OF VIBRATION DAMPING AND ACOUSTIC

ATTENUATION IN AEROSPACE TECHNOLOGY

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Application of damping treatments used in the aerospace industry will be illustrated in this paper by the work of the General Electric Company, Space Division, in applying its syntactic epoxy foam material to various spacecraft equipments. The material, dubbed SMRD after the Spacecraft Materials Research and Development organization, is a very effective damping material for this type of application. It is a proprietary, flexible epoxy compound containing silica microspheres and, as formulated for aerospace use, a certain amount of air bubbles. A number of different SMRD formulations are available. Figure 1 illustrates damping effectiveness vs. temperature for two of the most commonly used formulations in a typical constrained layer configuration. Figure 2 shows before and after response in a similar configuration. Table 1 lists a number of actual applications of SMRD damping treatment with before and after results. It is clear from inspection of the table that some remarkably effective damping treatments have been produced by a combination of an effective material and good dynamic analysis and design.

Computer programs have been developed and verified experimentally which accurately predict the stiffness and damping effects of the SMRD material applications. The SMRD material properties are used to calculate the composite loss factors (damping) and the resonant frequencies of the configuration. The programs have been used effectively in the applications of the damping material to electronic packages and have predicted the dynamic performance of such designs as the steel bars, the IUE camera bracket, the IUE camera deck, the Shuttle Payload Acoustic Cover and many printed circuit board applications.

Careful analysis is required in order to obtain good results since the material is non-linear and strain-rate sensitive. The design configuration is highly dependent on the particular requirements of each application. Analysis is required to properly size and position the damping material in relation to the rest of the structure so that optimum energy dissipation in the material can be achieved while meeting frequency, alignment, and strength

requirements. The SMRD damping material can make a significant stiffness contribution to a design.

Initial applications of damping material were for the control of electronic part vibration in LANDSAT 1 components (1). Subsequently, a number of research and development programs and contracts have been used to develop further and apply damping materials (2,3,4). This experience spans a variety of applications ranging from the addition of damping to resonant portions of existing structures to the design of integrally damped structures.

Table 1 lists some examples with results obtained in experimentally verified applications to spacecraft flight hardware and in development projects. A few LANDSAT component examples are given in which dynamic loading as measured by the reduction in magnification factor is quite dramatic. Similar results in random vibration were also obtained for the AMS mount hardware which utilized a SMRD truss stiffener. Equally good results were obtained with Viking Program components, GE's Sonar Transducer, and the other research elements which are included. All of these were in the "add-on" category. For the IUE brackets and mounts and the Shuttle Payload Acoustic cover, however, the SMRD was an integral part of the design. With such a "built-in" approach, much lower magnification factors were obtained and with optimized weight control.

Early vibration development tests of the NASA-Goddard Space Flight Center LANDSAT Spacecraft, Power Switching Module (PSM) indicated that, although the packaging design provided adequate stiffness, additional damping was needed to assure the adequacy of the design. The component vibration specification subjected the PSM to sinusoidal vibration levels in the thrust axis of 10g in the frequency range from 60 to 150 Hz. Consequently, the design goal was to provide resonances above 150 Hz. The PSM was comprised of two major sections; a section containing printed circuit boards and a section containing two relatively large relay panels (Figure 3). Although adequate stiffness could readily be provided for the PC boards, the relay panels weighed approximately three pounds each and were designed using 0.093 inch aluminum sheet supported to the basic box structure on three edges and included an angle stiffener in the center of the panel.

The vibration response near the center of the panel, showed that the fundamental resonant frequency was adequate, 160 Hz, but that the dynamic magnification, 40, was excessive. A SMRD 100F50 combination stiffening and damping treatment was applied to the PSM relay panels. The center stiffening angle was removed from the panel and the SMRD layer cast on the inner side of the panels was machined to

the specified thickness. SMRD 100F50 pads, machined to the correct thickness, were also bonded to the surface. Vibration tests of the modified relay panel installed in the PSM showed that the layered damping treatment reduced dynamic magnification of 40 in the panel center to approximately six (more than a factor six, Figure 3). Similar reductions were also obtained for the high frequency resonance between 500 and 1000 Hz. The fundamental panel resonance was increased slightly from 160 to 180 Hz which was sufficient to remove the resonance from the severe vibration environment below 150 Hz. This spaced damping treatment was highly effective in controlling the relay panel vibration levels.

SMRD 100F50 was also used to reduce the resonant amplitudes of a Hamilton-Standard all-metal mount for a strapped-down Inertial Reference Unit (IRU) for the Viking Program. In order to preclude misalignments resulting from hysteresis in conventional elastomeric vibration isolators, an isolator using metal as the main support was designed (Figure 4). The flexibility is provided by four right angle beam members that supported the unit to the mounting surface. However, at the isolator resonance of approximately 200 Hz, a dynamic amplification of 50 occurred, which was unacceptable. The size of the beam members was reduced slightly and a shear layer of SMRD 100F50 was applied between the beam members and an outer shear plate (Figure 4). With this modification of the mount, the resonant amplification was reduced to 2 1/2 to 4 in its major resonance.

The reduction of the vibration levels of the Printed Circuit Boards of the Martin-Marietta DAPU demonstrated the effectiveness of the damping material as a shear layer. The DAPU consists of a series of metal picture frame "slices" each containing two large PC boards as shown in Figure 5. The metal slices were bolted together through the four corners and at intermediate points along the edges. Adjacent slices had conformal coating on one pair of boards (slice 4) and SMRD 100F90 damping strips between another pair of boards (slice 5). The measured transmissibility plots are shown in Figure 6 for the center of the two boards. The fundamental board resonance between 200 and 300 Hz is evident in both plots. The amplification of the fundamental resonance was reduced from 60 for the conformally coated board to two for the SMRD damped board. The high damping of the SMRD board is evident from the large bandwidth of the resonant peak. At higher frequencies, resonances of the box structure make the damping somewhat less effective, but the damping still limits the amplification to approximately 8 as compared to 25 for the higher modes of the conformally coated board.

The Space Shuttle program presents new challenges in acoustic attenuation. The general arrangement of the Shuttle places the payload close to the rocket engine exhaust, which results in a more severe acoustic environment than that of current launch vehicles. A design goal for the Shuttle has been to limit the acoustic environment within the payload bay to 145 dB. However, current estimates of the attenuation provided by the Shuttle structure indicate that the levels are more likely to be about 150 dB as shown in Figure 7 (5,6,7,8). The Shuttle structure must provide more attenuation than that obtained with smaller diameter launch vehicle shrouds as indicated in the figure. On the other hand, continuing efforts to minimize the structural weight of the Orbiter has resulted in numerous design changes not yet factored into acoustic predictions. One of the more recent changes was to use lightweight graphite composite honeycomb doors. This weight reduction will probably reduce the acoustic attenuation provided by the Orbiter structure, as will the effects of vents and door seals which have not yet been included. To further complicate the predictions, the results of recent acoustic model tests at NASA-MSFC have indicated that the noise levels in the low-frequency acoustic environment within the Shuttle payload bay will be higher than the 145 dB Shuttle design goal and will be considerably more severe than that of current launch vehicles, particularly in the low frequency range (4).

One approach to protecting the payload experiments from the high acoustic levels predicted for the Shuttle is to enclose them in a lightweight damping canister. Such a canister, if it can be made efficient enough, will be cost effective in that many current items of equipment will not have to be redesigned and requalified for the higher Shuttle acoustic environment. An analytical and experimental study was done at GE to determine the acoustic performance of a canister. A cylindrical canister was designed, analyzed, manufactured, and tested in this program for NASA-Goddard Spaceflight Center (4).

The enclosure configuration consisted of two face sheets joined by a viscoelastic shear layer. Figure 8 shows the cylinder wall construction which was optimized using GE computer codes to provide high stiffness and damping. The complete assembled enclosure is shown in Figure 9. Measured noise reduction during testing, at the NASA-Goddard Space Flight Center, showed excellent agreement with theoretical noise reductions. Figures 10 and 11 summarize the noise reduction results of the enclosure without and with an acoustic liner, respectively. Figure 12 shows predicted Shuttle noise levels with various canisters (9). Based on the results of the study the following conclusions are made:

- Lightweight acoustic enclosures (approximately one third that of launch vehicle shrouds) are feasible to protect the Shuttle experiments.
- The test enclosure provided an order of magnitude reduction in the acoustic levels.
- Measured noise reduction agrees well with theoretical behavior.
- There was no apparent damage to the test enclosure after approximately 1/2 hour of exposure to acoustic levels from 146 to 154 dB.
- An internal sound absorbing liner should be provided within the enclosure.
- Acoustic enclosures with surface densities of 0.25 to 0.5 lbs/ft² are feasible.
- Larger covers are feasible.

CONCLUSION

High loss damping materials combined with insightful design and analysis can lead to practical and efficient damped structures.

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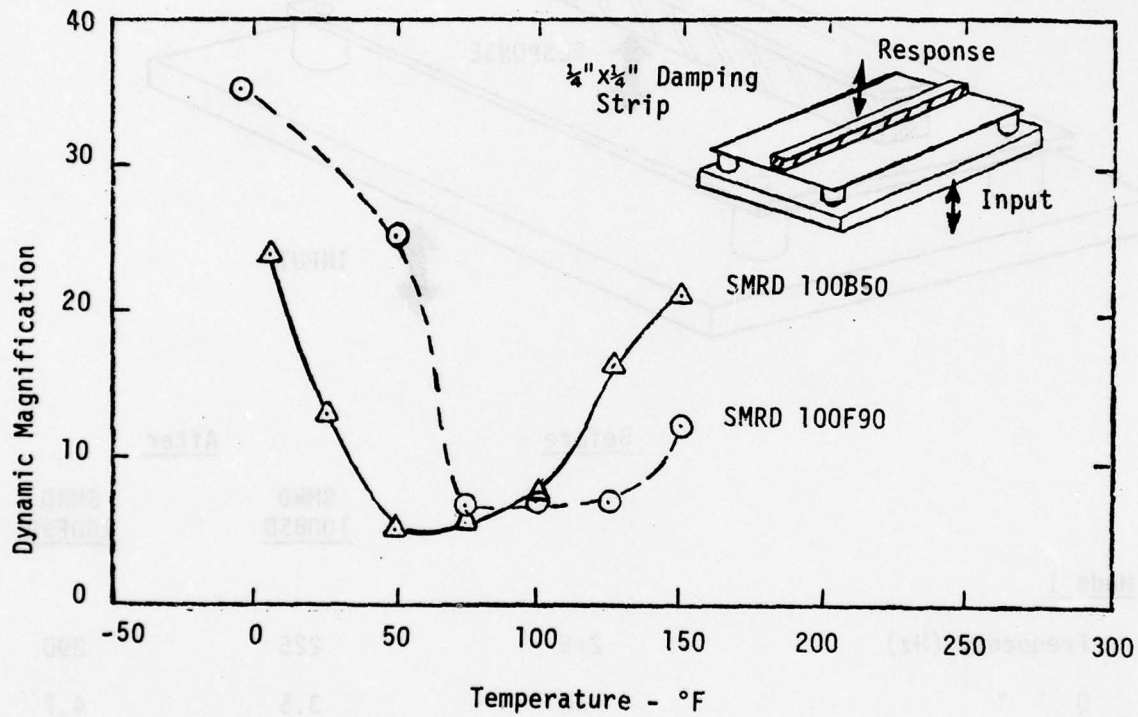
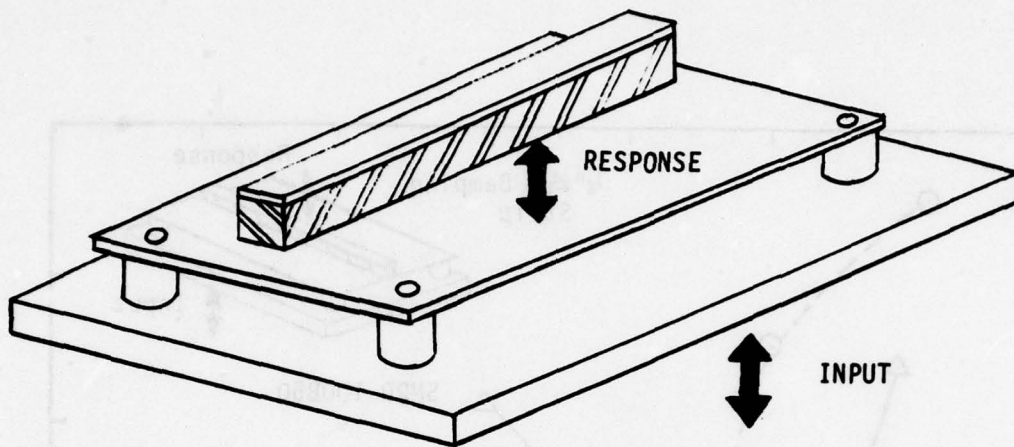


Figure 1. Printed Circuit Board Response with Epoxy Damping Strip of Two Different Materials



	<u>Before</u>	<u>SMRD 100BSD</u>	<u>After</u> <u>SMRD 100F90</u>
<u>Mode I</u>			
Frequency (Hz)	215	225	390
Q	38	3.5	4.7
<u>Mode II</u>			
Frequency (Hz)	1160	1150	1650
Q	29	1.3	1.5

Figure 2. Effect of Damping Strip on Response of Single PC Board (1/4" x 1/4" SMRD Strip)

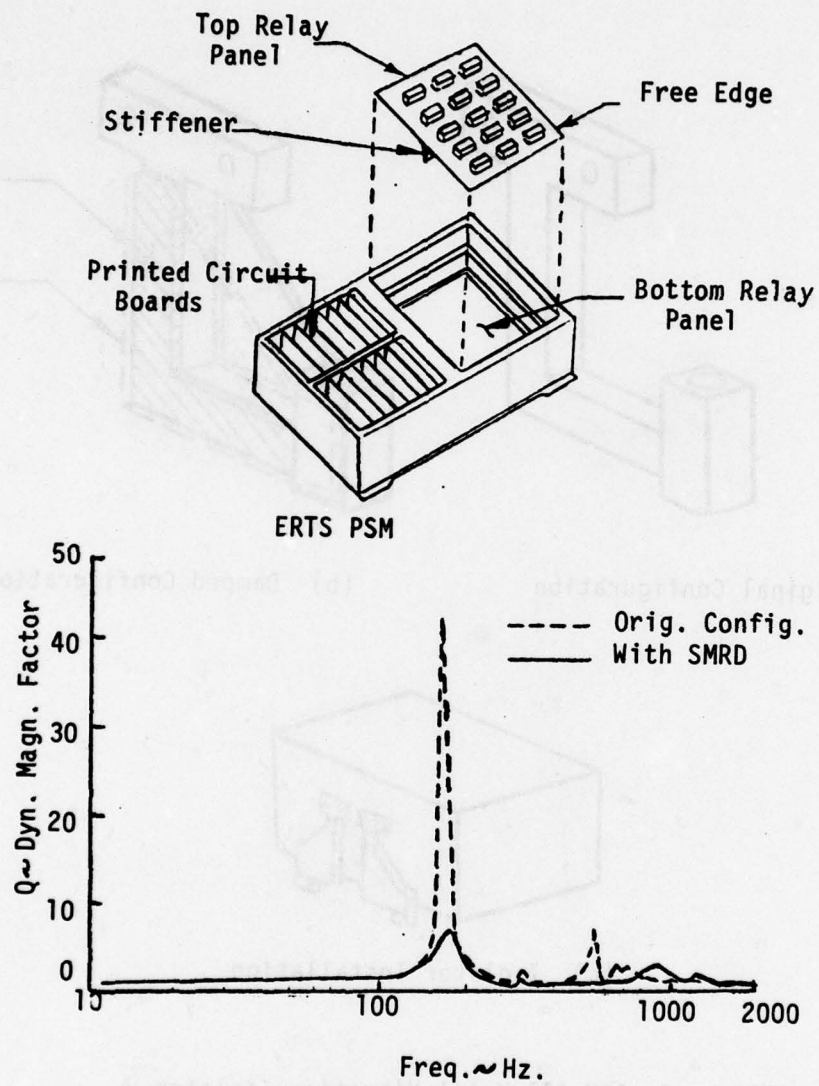


Figure 3. LANDSAT-PSM Relay Panel

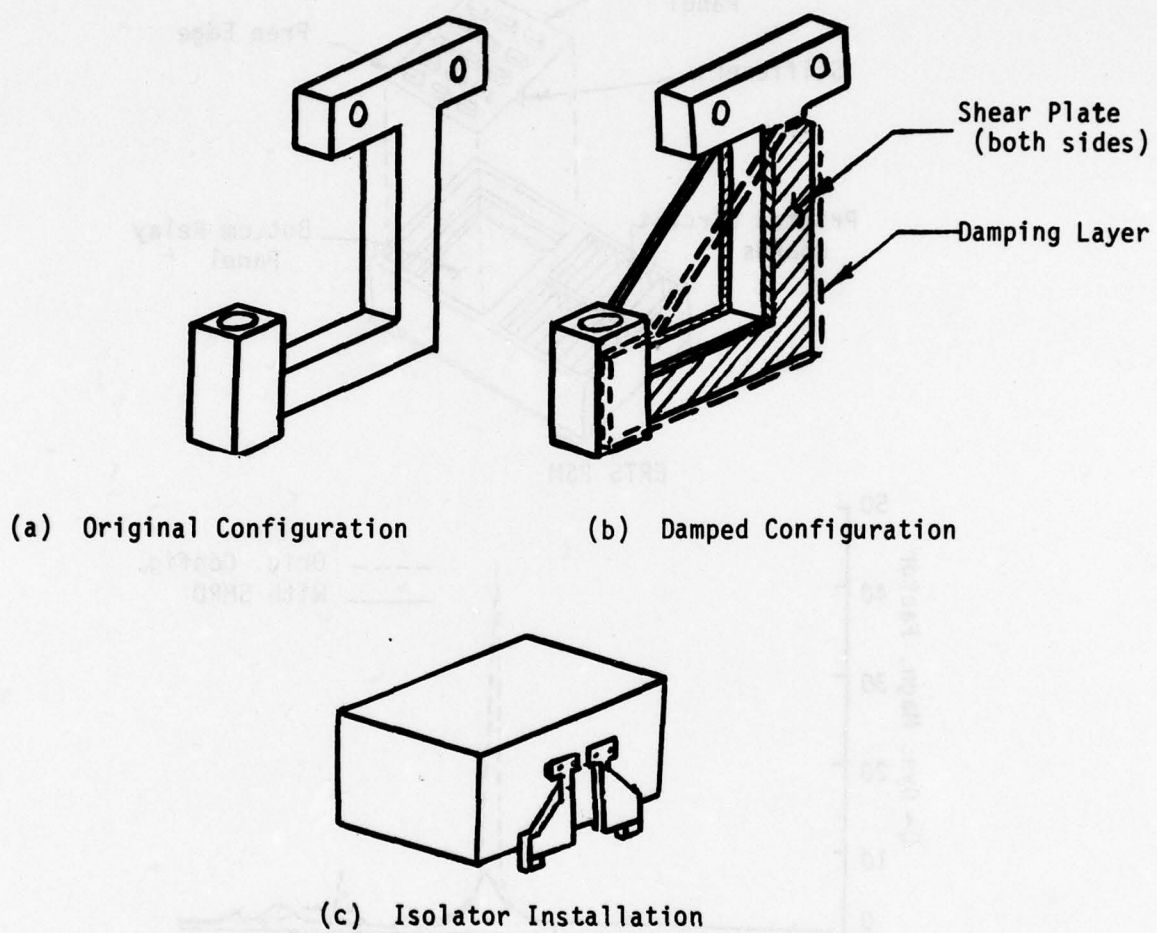


Figure 4. IRU All Metal Vibration Isolator

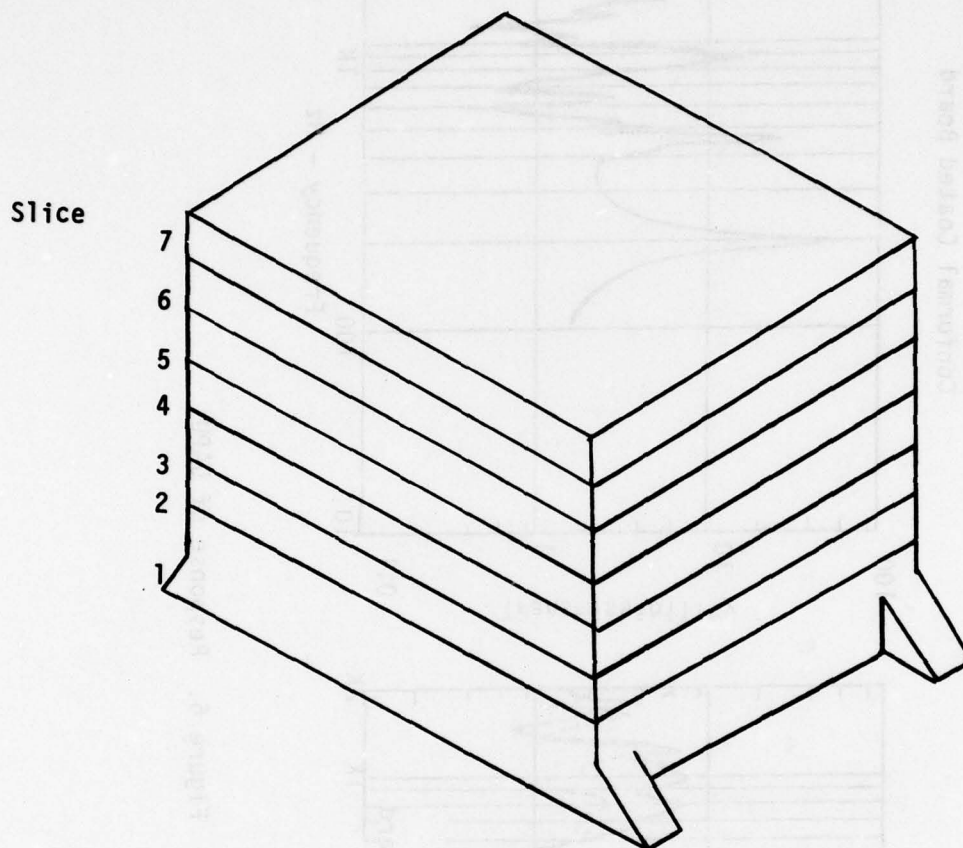


Figure 5. DAPU Configuration

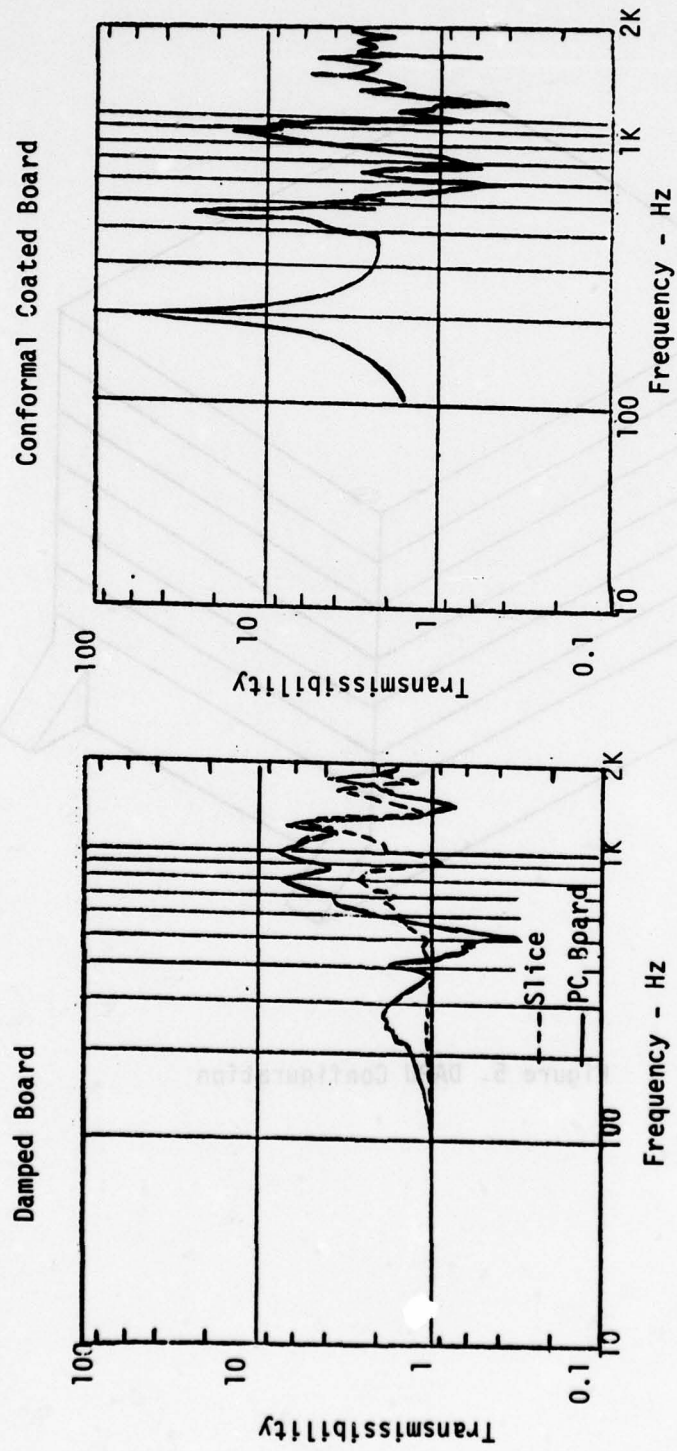


Figure 6. Response of DAPU

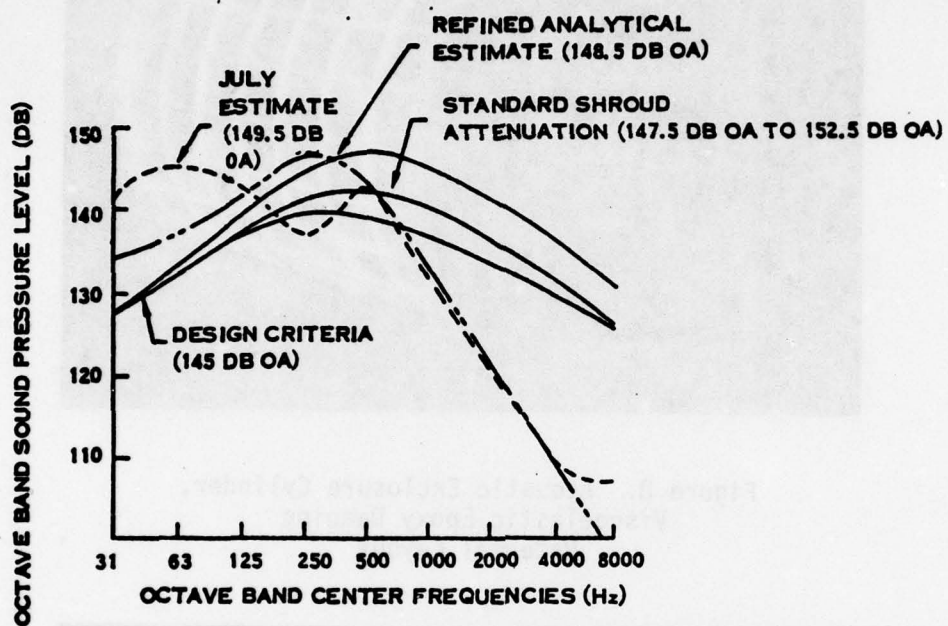


Figure 7. Internal Payload Bay Acoustic Spectra

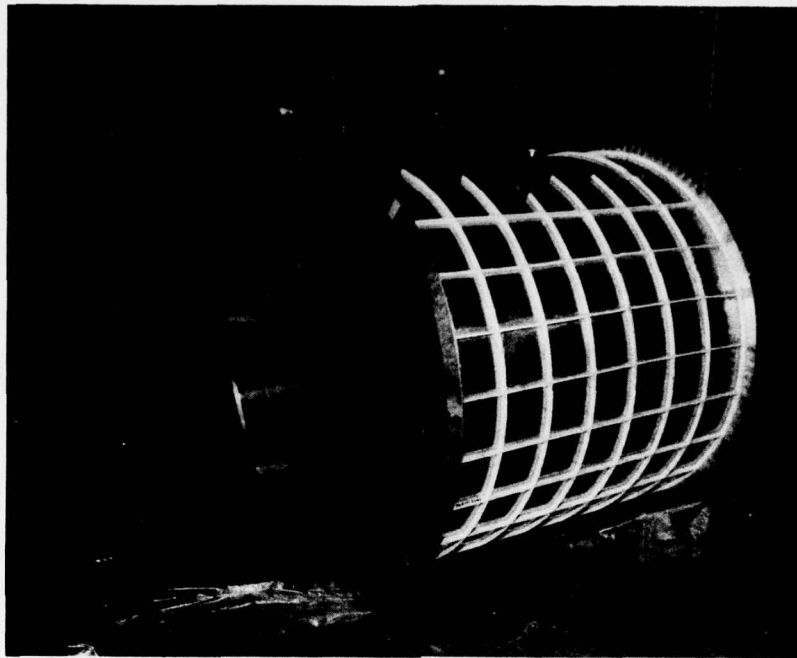


Figure 8. Acoustic Enclosure Cylinder,
Viscoelastic Epoxy Damping
Material Layout

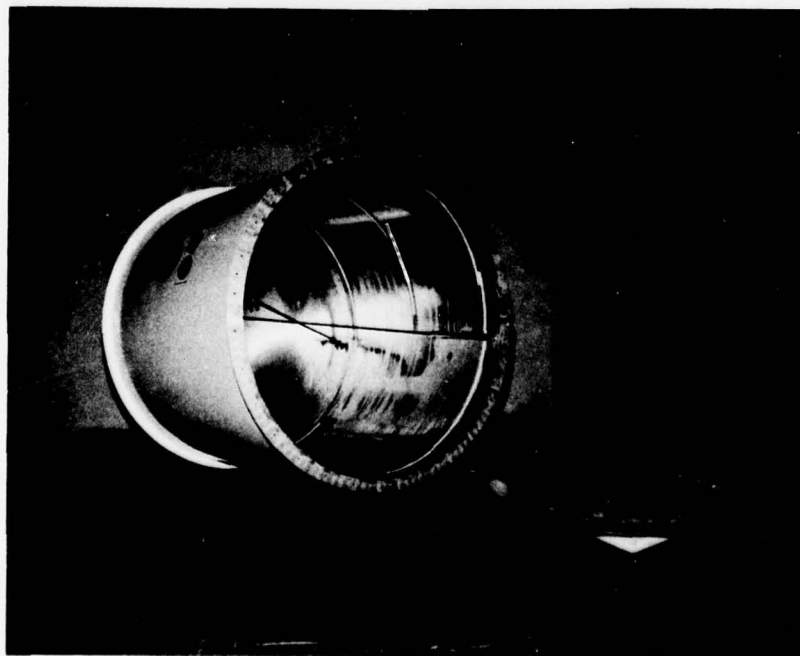


Figure 9. Acoustic Enclosure, One Bulkhead
Removed with the Fiberglass Liner

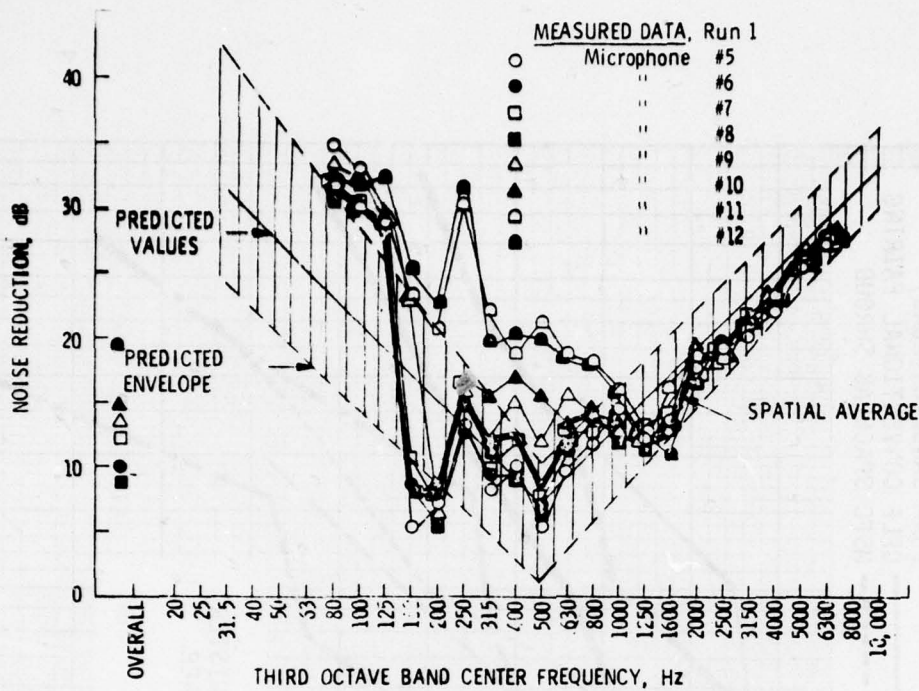


Figure 10. Acoustic Enclosure, Measured Noise Reduction Comparison to Predicted Values; Without Acoustic Fiberglass Liner (OA external SPL = 150.5 dB)

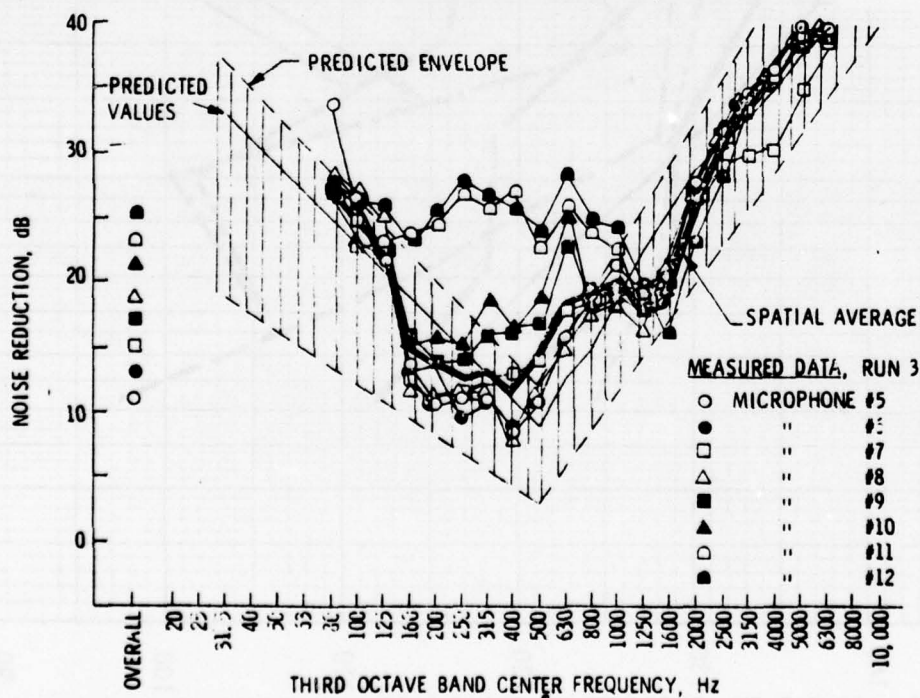


Figure 11. Acoustic Cover, Measured Noise Reduction Comparison to Predicted Values; With Acoustic Liner (OA external SPL = 150 dB)

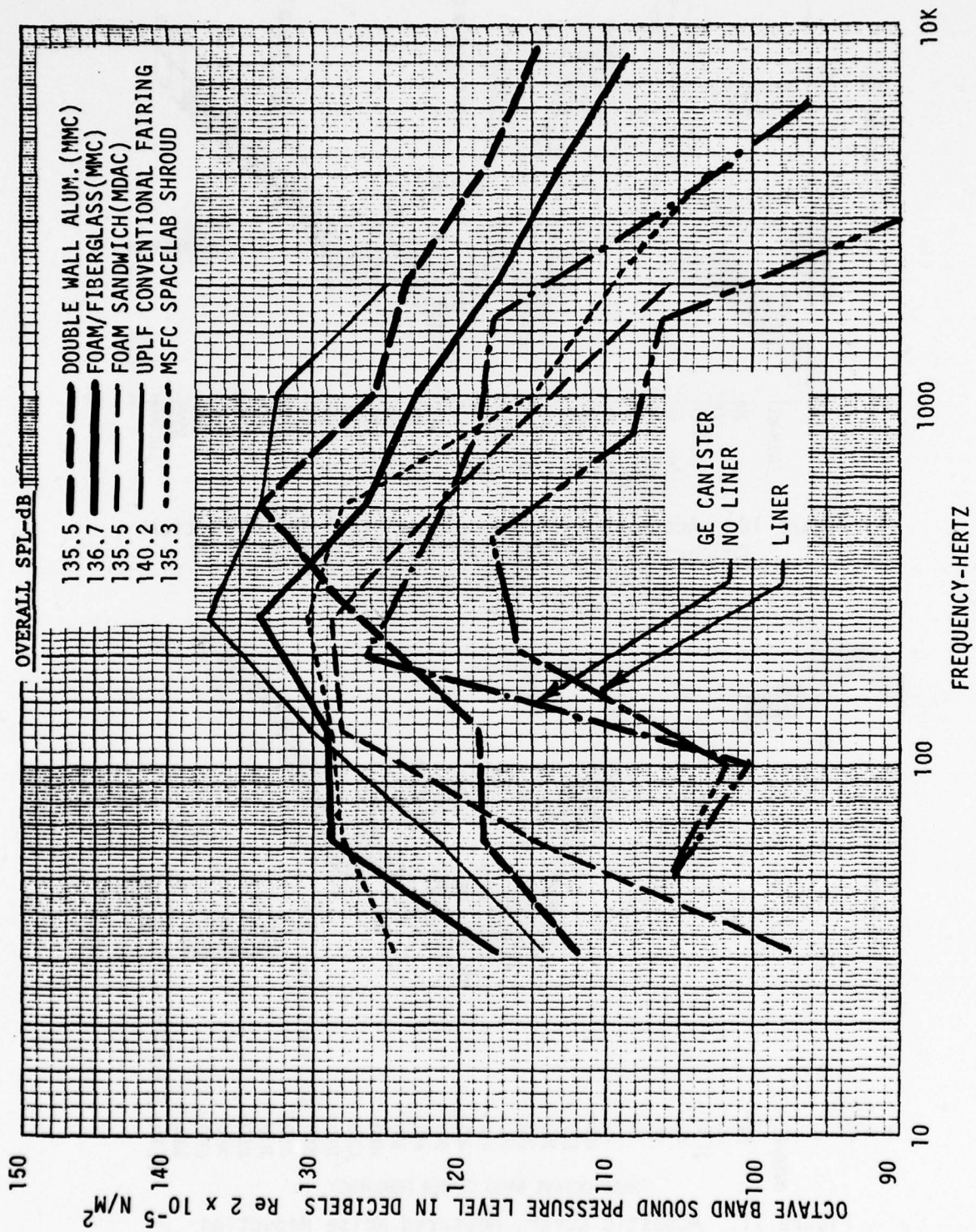


Figure 12. Predicted Payload Sound Pressure Levels Based Upon JSC 0700 Acoustic Levels

TABLE 1

SUMMARY OF TYPICAL SMRD DAMPING APPLICATIONS

	Amplification (Q)	
	Before	After
ERTS PSM Relay Panel (GE-SD)	40 @ 160 Hz	6 @ 180 Hz
ERTS AMS (Quantic)	44 @ 675 Hz	16 @ 1950 Hz
ERTS WBFM (GE-SD)	58 @ 480 Hz	7.6 @ 1600 Hz
Viking DAPU (Martin Marietta)	50 @ 190 Hz	3 to 6 @ 320 Hz
Viking IRU - IDD** (HAM-STD)	50 @ 200 Hz	2 1/2 to 4 @ 200 Hz
Sonar Transducer (GE-HMES)	10 @ 700 Hz	1.3 @ 1000 Hz*
PC BRD Damping Strip (Martin Marietta)	45 @ 230 Hz	3.2 @ 275 Hz
Steel Bar Comparison (GE-IR&D)	44.4 @ 15 Hz 49.2 @ 100 Hz 39.6 @ 280 Hz	6 @ 48 Hz 2.4 @ 230 Hz 15. @ 550 Hz
20" Dia. Bulkhead - IDD** (GE-IR&D)	65 @ 400, 1800, 3500 Hz	2.5 @ 350, 1500, 3000 Hz
20" Dia. Bulkhead with 50 lb. mass - IDD** (GE-IR&D)	40 @ 90 Hz	2.0 @ 60 Hz
IUE Camera Bracket - IDD** (NASA-GSFC)	15 to 20 @ 100 to 2000 Hz (several modes)	1.5 to 5 @ 77 to 400 Hz (isolation over 450 Hz)
IUE Camera Deck (NASA-GSFC)	7 to 35 @ 80 to 2000 Hz (several modes)	1.4 to 4.8 at 80 to 2000 Hz
Shuttle Payload Acoustic Cover - IDD** (Reference 4)	(Shuttle Payload Env. simulated at 150 dB SPL overall applied external to cover)	Noise reduction (measured inside cover) Low freq. - 30 dB Mid freq. - 15 dB High freq. - 35 dB Overall - 20 dB

*Predicted Based on Material Test Results

**Integrally Damped Design

DESCRIPTION AND MECHANICAL PROPERTIES OF A
PRE-PRODUCTION COMPOSITE STRUCTURE

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To meet the modern thrust of increased demands on conventional materials, the area of composites has evolved rapidly and is considered by many to be the predominant outlet for new materials of the future. A composite material is a combination of two or more materials with different properties that yield unique properties unattainable in either component alone. The composite structure that I will describe in this talk is a classic example of this phenomenon.

The intensive study of composite materials in recent years has been primarily motivated by the desire to produce improved materials for specific applications which require a high strength- or stiffness-to-weight ratio. Of course, these are not the only requirements; however, they are the ones which are most amenable to the experimental and analytical techniques of applied mechanics and will be exclusively considered in this talk. In particular I will be concerned with understanding the behavior of a relatively new engineering material, syntactic foam.

Syntactic foam is a two-phase composite material consisting of hollow microspheres integrally cast in a resin matrix. The word syntactic comes from the Greek prefix "syn" meaning with or together and the noun "taxis" meaning an orderly division or arrangement. Thus, syntactic means "put together in an orderly arrangement" and hence syntactic foam is an orderly arrangement of microspheres in a resin matrix. Such a foam is in contrast to other foams which are made by creating cavities in a single material. For our purpose the greatest interest is centered around those utilizing glass spheres with diameters in the range of 20 to 80 microns (0.0008 to 0.0032 in.) and with wall thicknesses of 1 to 3 percent of their diameters. In the dry, non-embedded condition, they constitute a semi-fluid aggregate having the appearance and consistency of a fine-grained sand whose mobility is limited by inter-granular friction only. Embedment in a resin matrix at a ratio of approximately two-thirds the composite volume transforms the aggregate into a solid which has finite strength under all loading conditions, and thus it becomes a useful structural material. In Figure 1 is shown a 100x photomicrograph of the structure of a typical syntactic foam. Since the

spheres are hollow, the resulting composite may have densities in the range of 35 to 50 lb/cu ft. The material thus has a remarkably high strength-to-weight ratio and has already received important applications as a buoyancy material for deep submergence vehicles, where depths of 25,000 to 40,000 feet can be attained before failure occurs. In addition, such composites can be used for energy-absorbing layers or as core materials for sandwich plate and shell constructions. No doubt they will find many other structural applications.

In the first section of this talk I will discuss the qualitative and quantitative mechanical behavior of syntactic foams. Following that, I will show that the micro-mechanical approach (which constructs idealized models of the microstructure and from the model and known constituent properties, attempts to predict overall composite behavior) does indeed produce very good estimates of the elastic properties of syntactic foam. Finally, I will present some specific results for a particular three-component composite foam and again show that for at least the static case, this approach relates reasonably well with experimental observations.

Figure 2 presents some general information regarding a general purpose syntactic foam that is commercially available from the 3M Company. A general description of the foam, several possible applications, available grades, and general properties are given. Note the low density, low water absorption, and low coefficient of thermal conductivity which are typical of what one might expect from such a composite.

Figure 3 presents more detail on some typical measured properties for two 3M syntactic foams of 42 pcf and 36 pcf. Note the high compressive and bulk modulus and lower shear modulus which are typical values for solid polymeric materials.

I will now briefly outline the micro-mechanical approach to predicting these properties. Recall that this method constructs an idealized model of the microstructure and from this and the known properties of the individual components predicts the elastic constants of the overall composite.

Much successful work in bulk- and Young's-modulus prediction for composite materials has been through the use of certain bounding techniques known as mixture theories. These techniques use variational theorems involving strain energy and work. In composite studies good results can be obtained using the relationships derived by Kerner. This method is particularly applicable when one of the materials

of the composite is a solid spherical inclusion. The Kerner predictions for the bulk and shear moduli are presented in Figure 4, where K_1 and K_2 , G_1 and G_2 , and V_1 and V_2 are the respective bulk-moduli, shear-moduli, and volume fractions of the components. The volume fraction of the glass microspheres is called the packing factor.

Figure 5 shows plots of the Kerner relationship for an EPON 828 standard resin. Assumed properties of the microsphere glass, high-strength resin, and EPON 828 or standard resin are presented in Figure 6. Actual property data for the microsphere glass are proprietary to 3M Company, whose microspheres are the most frequently used. The properties can be reasonably approximated for a soda lime borosilicate glass of which the microspheres are known to consist.

Figure 7 presents the effect of varying Poisson's ratio for a standard resin. The effect is small in the area of interest from 28 to 42 lb/cu ft foams, because of the decreased amount of resin with higher packing factors.

The Kerner method, as previously mentioned, is applicable to a composite in which one of the components is a solid spherical inclusion. However, it is obvious that the glass microspheres in a foam are not solid but instead are hollow. The bulk modulus of a hollow sphere is then given by the last equation in Figure 4, where "A" is the ratio of inside sphere radius to outside sphere radius, and γ is a material parameter equal to

$$\frac{2(1 - 2\nu_g)}{1 + \nu_g},$$

and ν_g is the Poisson's ratio of the glass.

Figure 8 presents the predicted Young's modulus for a standard resin. We see in Figure 9 good agreement between the measured values and those predicted. Because this method appears to correctly predict the elastic moduli, an empirical method for predicting the hydrostatic collapse strength is also available. These results are also shown in Figure 9, however, the details of this method are beyond the scope of this talk.

The next subject I will address is the high frequency sound or ultrasonic attenuation and velocity properties of syntactic foam. We all realize the importance of these ultrasonic properties in NDT work which is applicable here to syntactic foam inspection and quality control. Figure 10 shows the ultrasonic attenuation factor versus frequency for syntactic foams of varying density. As observed, the

attenuation increases with frequency for all three foams up to a frequency of 2.5 MHz and subsequently levels off. Figure 11 shows the relationship between the attenuation and the glass microsphere-resin content. We note that the attenuation is linearly related to the resin content. Finally, Figure 12 shows the effect of frequency on the sound velocity for one particular foam density. No apparent dependency of sound velocity on frequency is observed.

Returning now to theory, I note that the sound velocity may be predicted using the Kerner method employing the relation:

$$C = \frac{(K + 4/3G)^{\frac{1}{2}}}{\rho}$$

When the proper moduli are inserted from Figures 4 and 6, a value of 2.57×10^5 cm/sec is obtained, which is in excellent agreement with the experimental value found in Figure 12. Figure 13 presents additional information on the increase of sound velocity with hydrostatic pressure (or depth).

I now turn to the particular composite shown in Figure 14 which is of concern to this workshop group. The composite which was generated by the 3M Company is basically an epoxy or syntactic foam matrix in which particles of a polymeric material are embedded. The exact constituents and the details of manufacturing are beyond the scope and purpose of this talk. The matrix core is covered with glass-fiber reinforced plastic (GRP) skins. The purpose of the GRP skins is to provide structural strength. Utilization of skins on both faces results in a structure similar to the strong honeycomb structure used in the aircraft industry. Figures 15 and 16 present relevant data of a typical GRP skin material available from the 3M Company. The principal strength requirements for this material are that it will have high compressive, tensile and shear strengths in order to remain intact under various loading conditions. The tests taken to date are: compression, tensile, water absorption, falling ball impact, and bending beam. Figures 17, 18 and 19 show the experimental test setups. Shown in the upper half of Figure 20 are data obtained from eight samples, while the lower half summarizes other data. Although the minimum specific goal was met for some of the samples (upper half of Figure 20) there still exists a need to (a) improve the strength properties and (b) reduce the specific gravity of the composite.

Finally, just as the epoxy microsphere syntactic foam can be considered a composite material whose effective elastic constants may be found from mixture theories, the elastic constants of the syntactic foam-polymer composite may also be calculated by the same procedure. Figure 21 shows the predicted variation of the bulk moduli versus volume fraction of inclusions for the composite and is in excellent agreement with the data of Figure 20.

In summary, therefore, I conclude that mixture theories can adequately predict the elastic properties of a wide variety of composite materials and are most appropriate to a wide spectrum of syntactic foam composites. In addition, areas for improvement of the particular syntactic foam composite presented in this paper are: improved strength, reduced weight, and although no data was given here, reduced cost in manufacturability.

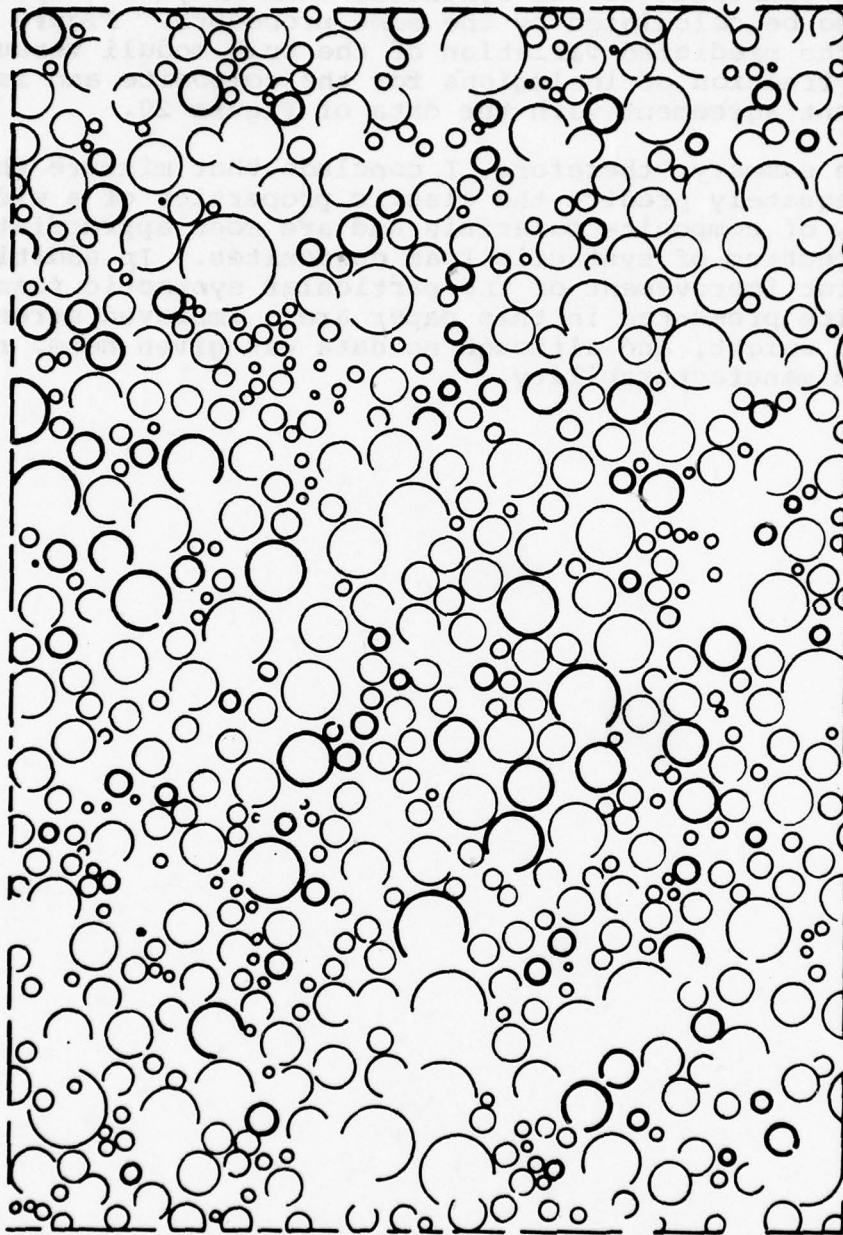


FIG. 1 100x PHOTOMICROGRAPH OF SYNTACTIC FOAM

"SCOTCHPLY" XP-241 SYNTACTIC FOAM

"SCOTCHPLY" XP-241 is an epoxy resin filled with hollow glass bubbles and cured to form a high strength, low density, hydrophobic foam.

Some possible applications follow:

- Applications which require a buoyant material with high strength and resistance to water absorption under high hydrostatic load.
- To prevent damage to instrument packages by hydrostatic forces experienced in deepsea oceanographic studies.
- A core material for sandwich construction.
- Provide buoyancy for deep diving submersibles.
- Acoustic windows for deep submergence application.

Available Grades of XP-241:

- 34, a nominal 34 pcf foam
- 36, a nominal 36 pcf foam, qualified to MIL-S-24154A, Type I
- 42, a nominal 42 pcf foam
- 42H, a nominal 42 pcf foam, qualified to MIL-S-24154A, Type II

General Properties: (Based on 42 pcf nominal density foam).

Density	42 \pm pcf from casting to casting
Water Absorption	Less than 3%w after 6 weeks at 10,000 psi on 1" dia. X 2" long cylinders.
Linear Cure Shrinkage	0.75 percent
Dielectric Constant @ 1MC	2.24
Dissipation Factor	0.77
Thermal Conductivity	0.0676 @ 90°F. BTU/hr./ft. ² /°F./ft. 0.0906 @ 180°F. 0.0980 @ 240°F.
Coefficient of Thermal Expansion	10.8 X 10 ⁻⁶ in./in./°F. (72°F. - 300°F.)
Izod Impact	0.13 ft. - lb./inch notch

FIGURE 2

COMPARED WITH MIL-S-24154A TEST VALUE REQUIREMENTS

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FIGURE 3

$$K_{\text{EFF}} = \frac{\frac{K_1 V_1}{3K_1 + 4G_1} + \frac{K_2 V_2}{3K_2 + 4G_1}}{\frac{V_1}{3K_1 + 4G_1} + \frac{V_2}{3K_2 + 4G_1}}$$

$$K_S = K_2 \frac{1 - A^3}{1 + \frac{A^3}{\gamma}}$$

FIGURE 4

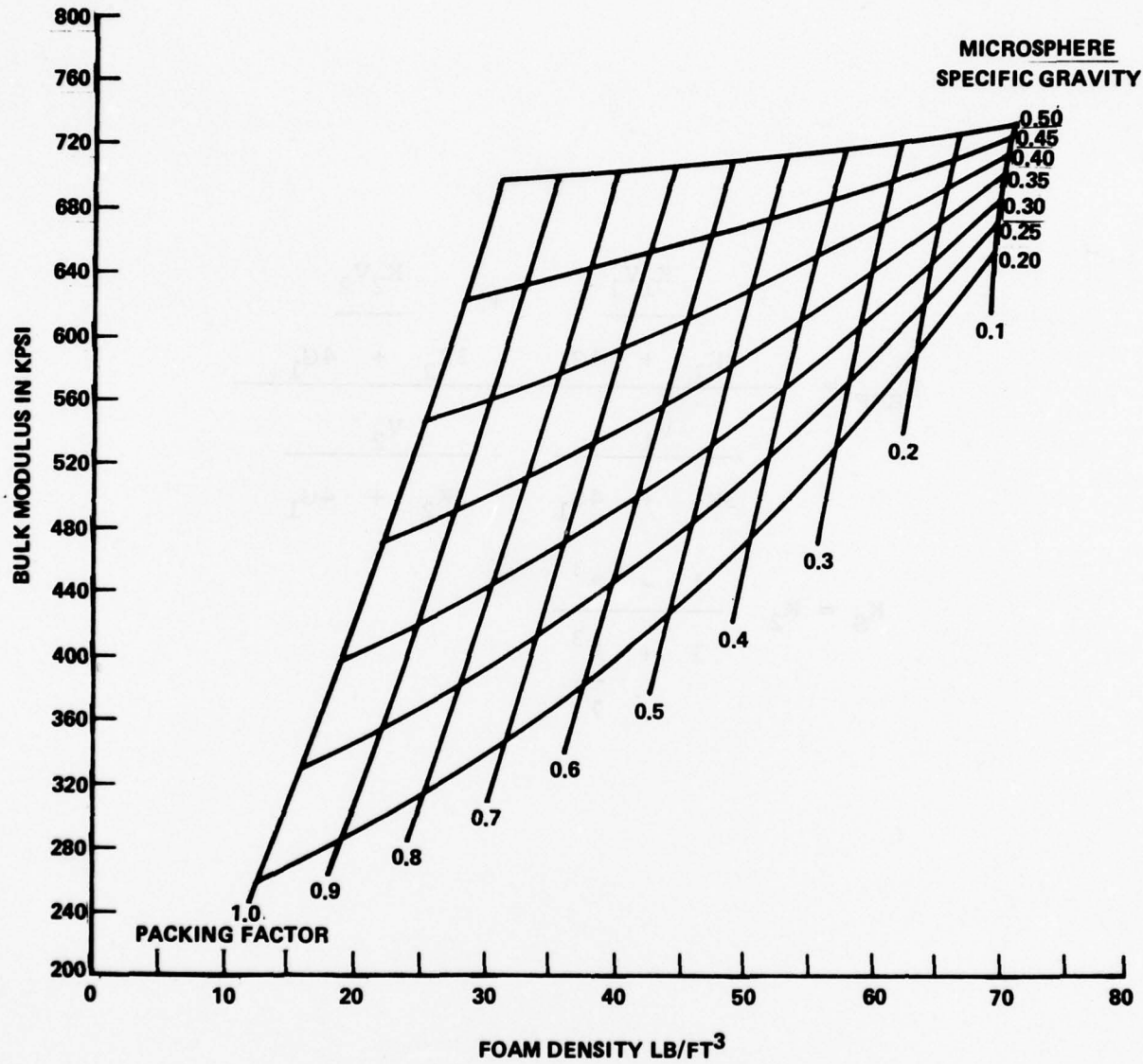


FIG. 5 BULK MODULUS FOR STANDARD RESIN SYSTEM

MATERIAL PROPERTIES OF SYNTACTIC FOAM COMPONENTS

Material	Specific Gravity	Young's Modulus (psi x 10 ⁶)	Poisson's Ratio
Standard strength resin	1.212	0.570	0.37
High-strength resin	1.26	0.850	0.37
Glass	2.37	10.40	0.21

FIGURE 6

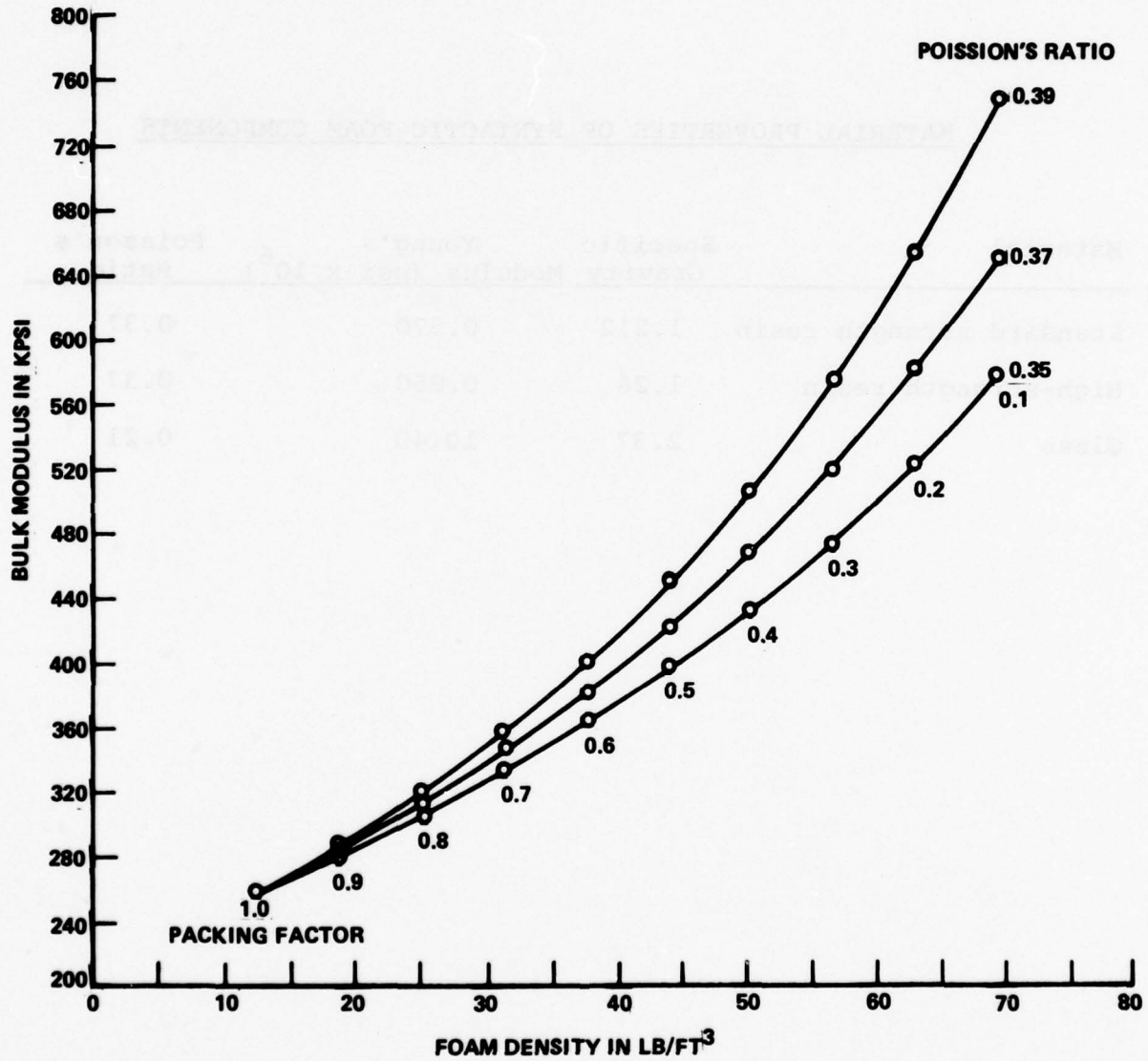


FIG. 7 BULK MODULUS OF VARYING RESIN, POISSON'S RATIO

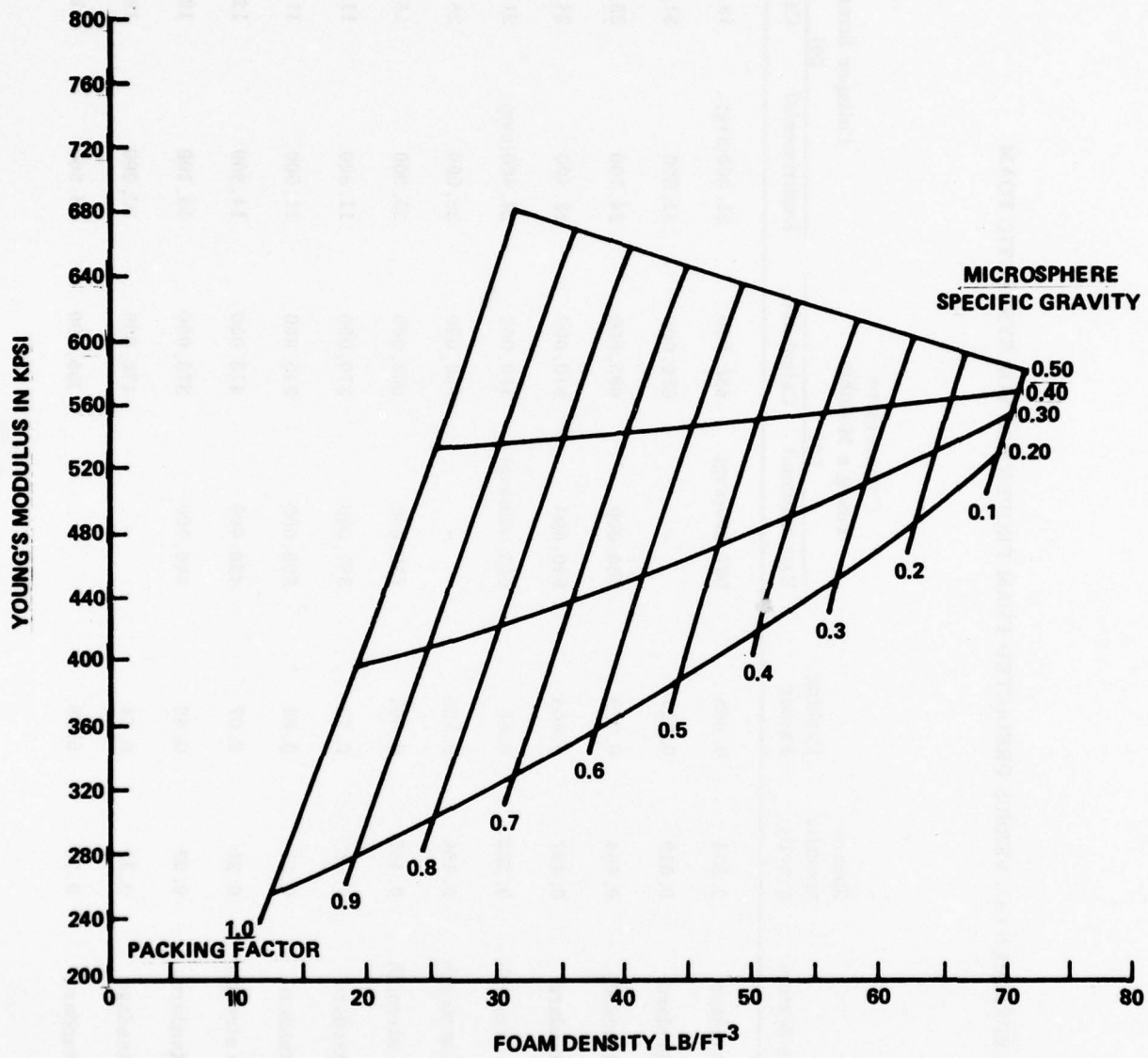


FIG. 8 PREDICTION OF YOUNG'S MODULUS FOR STANDARD RESIN SYSTEM

EXPERIMENTAL VERSUS PREDICTED FOAM PROPERTIES FOR SYNTACTIC FOAM

Foam Designation	Resin System	Sphere specific gravity	Packing Factor	Compressive Young's Modulus psi		Collapse Strength psi	
				Experimental	Calculated	Experimental	Calculated
NASL A, B, C	Standard	0.334	0.665	507,000(avg)	480,000	21,300(avg)	18,600
G*	Standard	0.519	0.617	-	650,000	31,500	31,800
H*	Standard	0.464	0.739	650,000	605,000	24,700	25,700
I*	Standard	0.467	0.804	640,000	610,000	23,400	24,800
J, K	High strength	0.334	0.66	645,000(avg)	560,000	24,400(avg)	21,400
P*	High strength	0.464	0.735	-	665,000	28,000	28,400
Q*	High strength	0.467	0.802	740,000	665,000	25,900	26,700
NSRDC 1	Standard	0.22	0.65	387,000	370,000	11,800	11,800
3	Standard	0.21	0.69	360,000	345,000	11,500	11,600
4	High strength	0.21	0.67	430,000	415,000	14,800	12,500
7	Standard	0.23	0.66	400,000	375,000	13,100	12,400
14	Standard	0.33	0.64	-	470,000	20,200	18,900
17	Standard	0.35	0.68	-	495,000	18,500	19,300

FIGURE 9

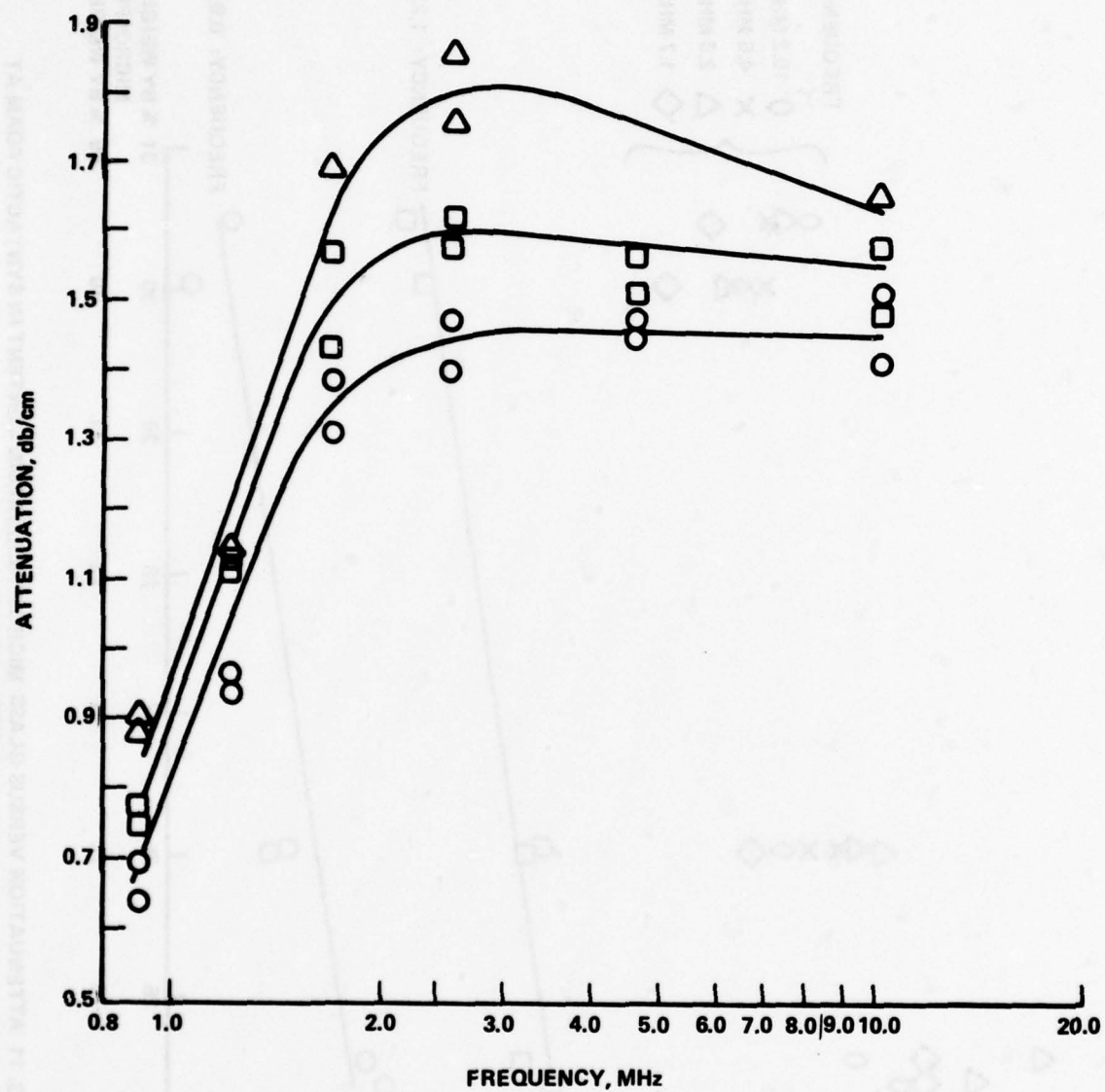


FIG. 10 ATTENUATION VERSUS FREQUENCY IN SYNTACTIC FOAM

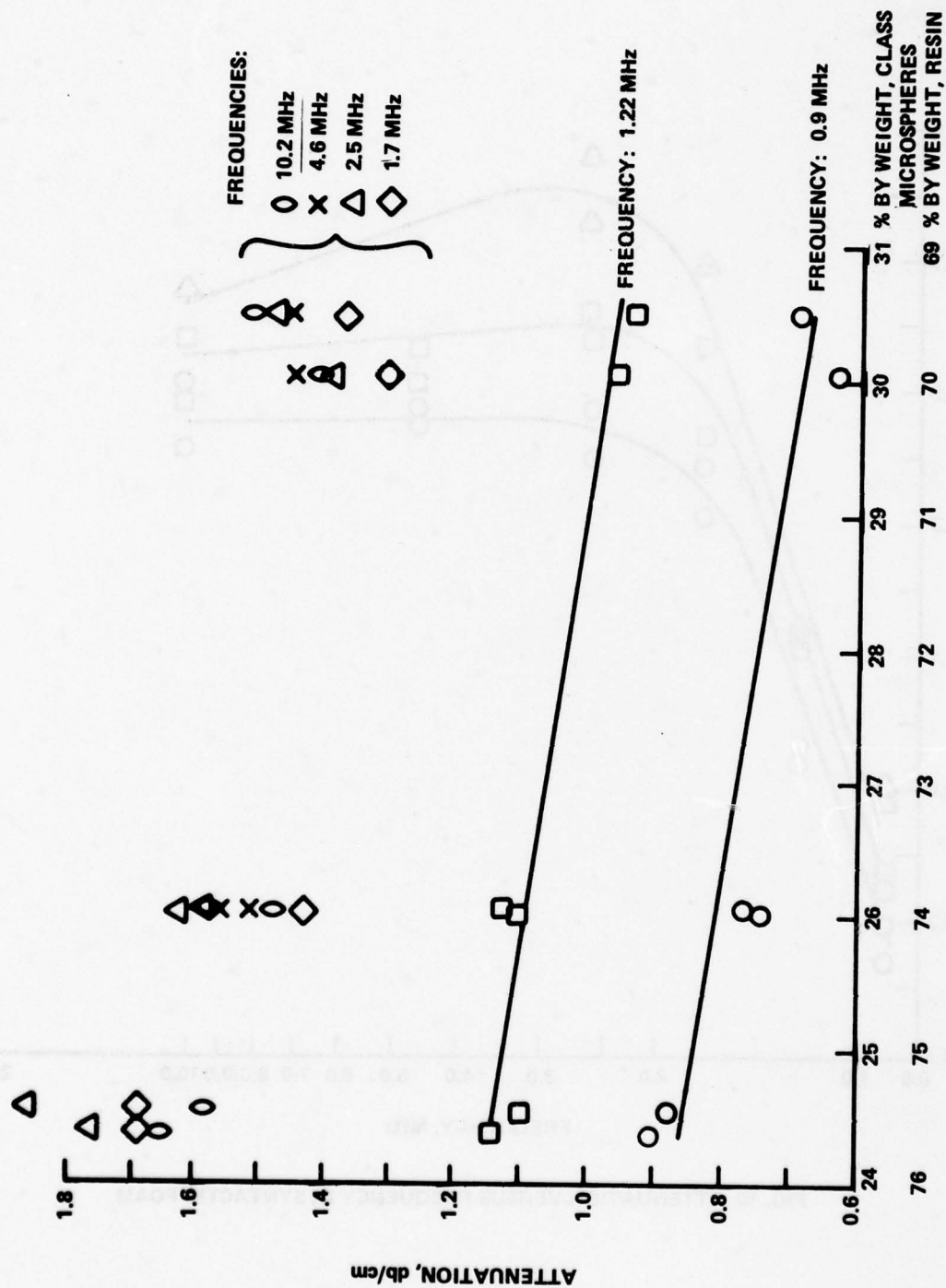


FIG. 11 ATTENUATION VERSUS GLASS MICROSPHERE/RESIN CONTENT IN SYNTACTIC FOAM AT DIFFERENT ULTRASONIC FREQUENCIES

ATTENUATION AND VELOCITY IN SYNTACTIC
FOAM AT DIFFERENT ULTRASONIC FREQUENCIES*

Frequency MHz	Attenuation ** db/cm	Velocity 10 ⁵ cm/sec	Frequency MHz	Attenuation ** db/cm	Velocity 10 ⁵ cm/sec
0.9	0.768	2.63	0.9	0.740	2.61
1.2	1.12	2.64	1.2	1.11	2.70
1.7	1.57	2.60	1.7	1.43	2.67
2.5	1.58	2.62	2.5	1.62	2.68
4.6	1.56	-	4.6	1.51	-
10.2	1.48	-	10.2	1.58	-

Density, pcf	42.10			41.74	
Glass Microspheres, % by weight :	26.05			26.01	
Resin, % by weight	73.95			73.99	

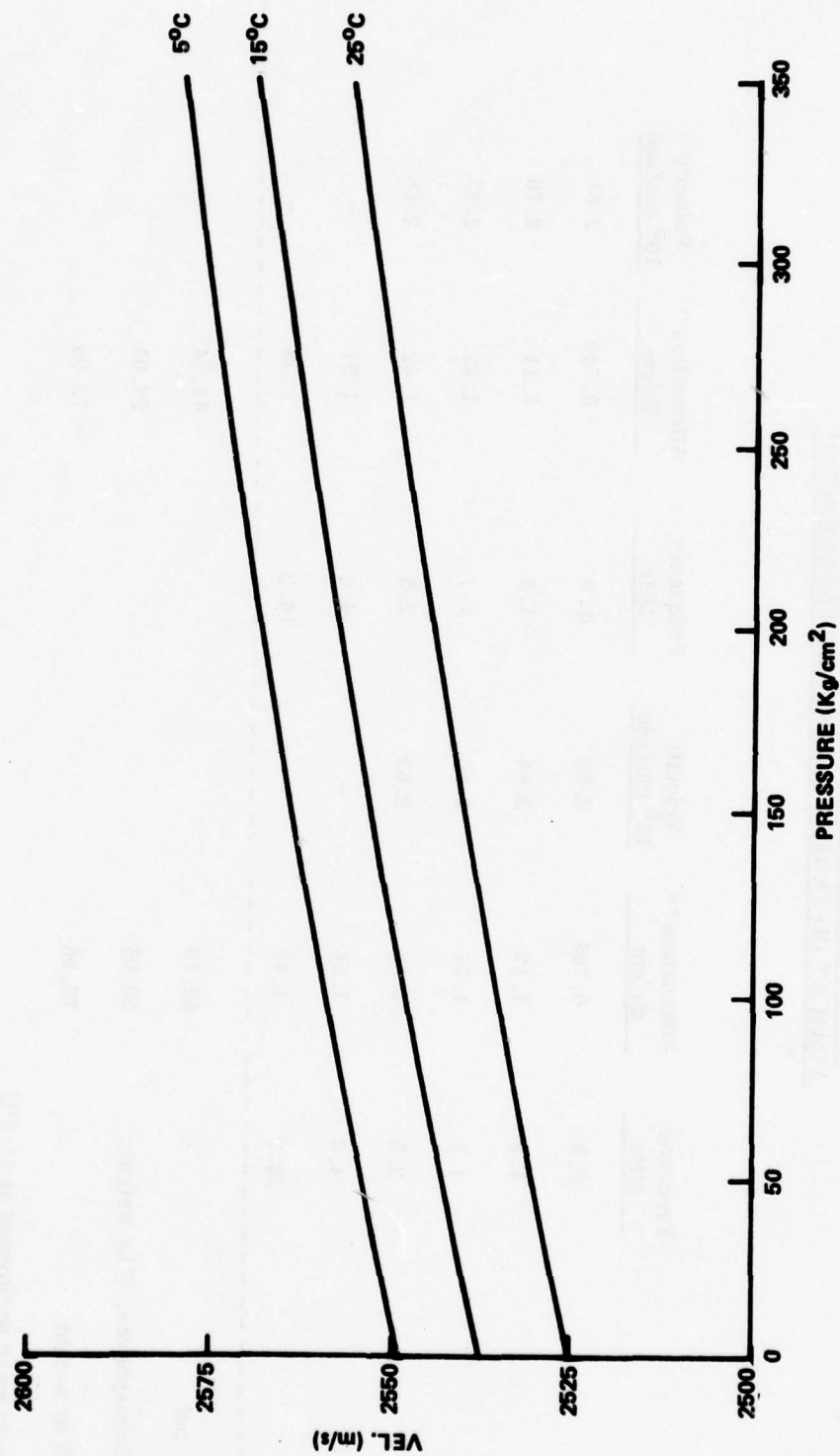
* - Tests were performed at 21.5°C

** - Attenuation corrected for surface reflectivity as per procedure in ref. (b)

*** - Specimen size : 3x3x9- $\frac{1}{2}$ inch (nominal)

$$C_L = \sqrt{\frac{K + 4/3 G}{\rho}} \qquad C_s = \sqrt{\frac{G}{\rho}} \qquad G = \frac{E}{2(1+\sigma)}$$

FIGURE 12

FIG. 13 SYNTACTIC FOAM 38 LB/FT³

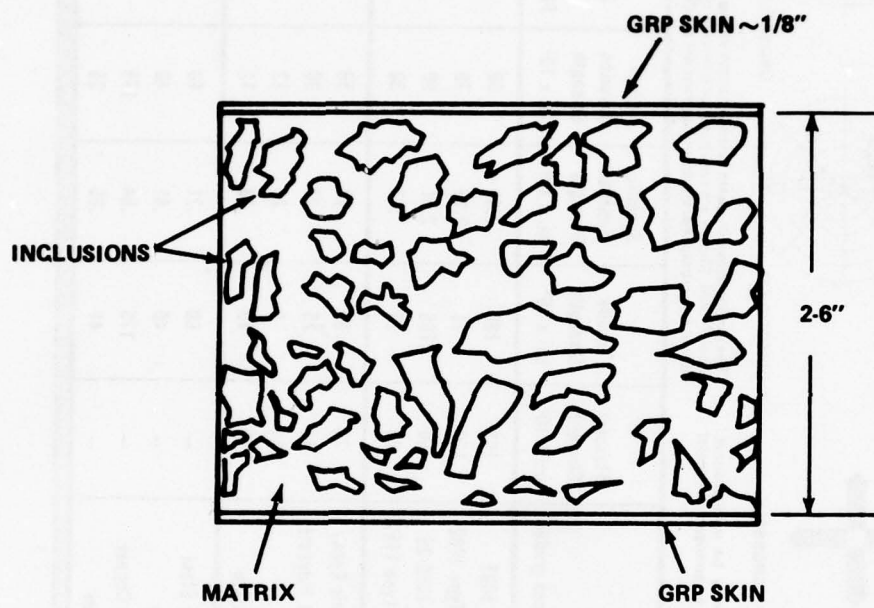


FIG. 14

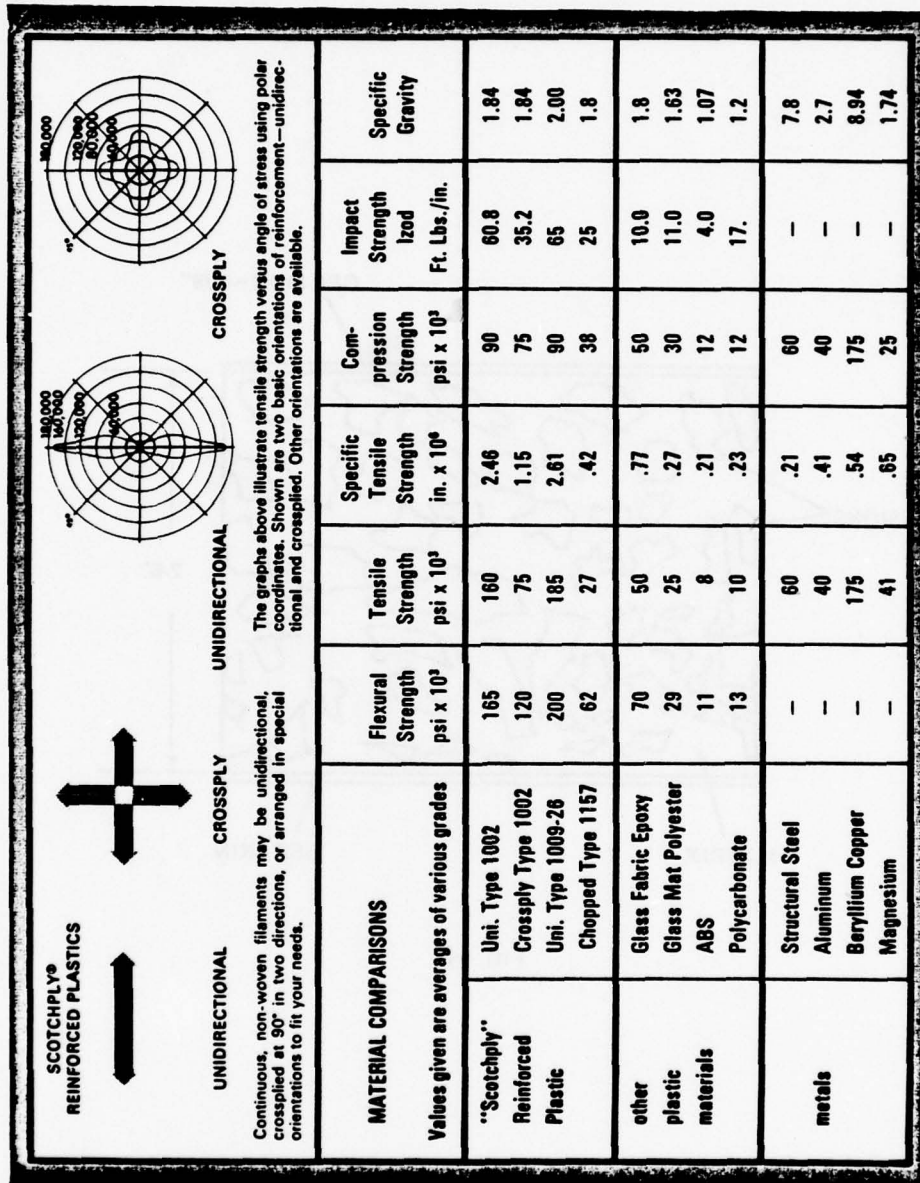


FIGURE 15.

PROPERTIES OF SCOTCHPLY® REINFORCED PLASTIC

	Units	1002		1008		1009-36		1009-26		1012		Molding Compound 1157
		Uni-direct'l	Cross-ply	Uni-direct'l	Cross-ply	Uni-direct'l	Cross-ply	Uni-direct'l	Cross-ply	Uni-direct'l	Cross-ply	
Tensile Strength	psi	160,000	75,000	155,000	75,000	160,000	75,000	185,000	72,000	140,000	72,000	27,000
Tensile Modulus	psi x 10 ⁶	5.7	3.7	5.7	3.7	5.8	3.7	5.7	3.5	5.7	3.5	3.0
Flexural Strength	psi	165,000	120,000	165,000	120,000	177,000	120,000	200,000	100,000	160,000	100,000	55,000
Flexural Modulus	psi x 10 ⁶	5.3	3.5	5.3	3.5	5.3	3.5	7.0	3.5	5.3	3.5	3.3
Compressive Strength (edge)	psi	90,000	75,000	80,000	77,000	87,000	77,000	90,000	76,000	93,000	76,000	40,000
Interlam. Shear Strength	psi	4,300	4,100	4,300	3,300	3,500	3,300	2,100	1,900	2,500	1,900	22,500
Izod Impact (edgewise)	ft. lb./in. of width	60	35	60	35	60	35	65	35	60	35	25
Barcol Hardness	-	70	70	70	72	72	72	74	70	70	70	70
Dielect. Strength (Cond. A)	VPM 1/16" thickness	750	750			650	650			630	630	450
Surface Resistance (Cond. A)	Megohms	4.0x10 ⁶	4.0x10 ⁶			2.7x10 ⁸	2.7x10 ⁸			1.8x10 ⁶	1.8x10 ⁶	1.1x10 ⁶
Vol. Resistivity (Cond. A)	Megohms cm.	6.6x10 ⁷	6.6x10 ⁷			1x10 ⁹	1x10 ⁹			1.6x10 ⁷	1.6x10 ⁷	6.5x10 ⁶
Dissipation Factor (Cond. A)	IMC	.017	.017			.010	.010			.010	.010	.011
Dielect. Constant (Cond. A)	IMC	4.8	4.8			5.2	5.2			5.2	5.2	4.8
Cured Thickness	in. per ply	.010	.020	.012	.020	.010	.020	.0075	.020	.010	.020	-
Specific Gravity	-	1.8	1.8	1.8	1.8	1.8	1.8			1.8	1.8	1.8
Resin Content	%	36	36	38	36	36	36	26	38	38	38	37
Max. Operating Temp.	°F	250	250	250	350	350	350	300	450	450	450	250
Water Absorption	%24 hr. immersion	.05	.05	.05	.05	.05	.05	.05	.05	.05	.05	.05
Wet Strength Retention	% 2 hr. boil	86	86	86	87	87	87	93	93	93	93	85
Flammability	-	Self Extinguishing	Self Extinguishing	Self Extinguishing	Self Extinguishing	Self Extinguishing	Self Extinguishing	Self Extinguishing	Self Extinguishing	Self Extinguishing	Self Extinguishing	-

FIGURE 16

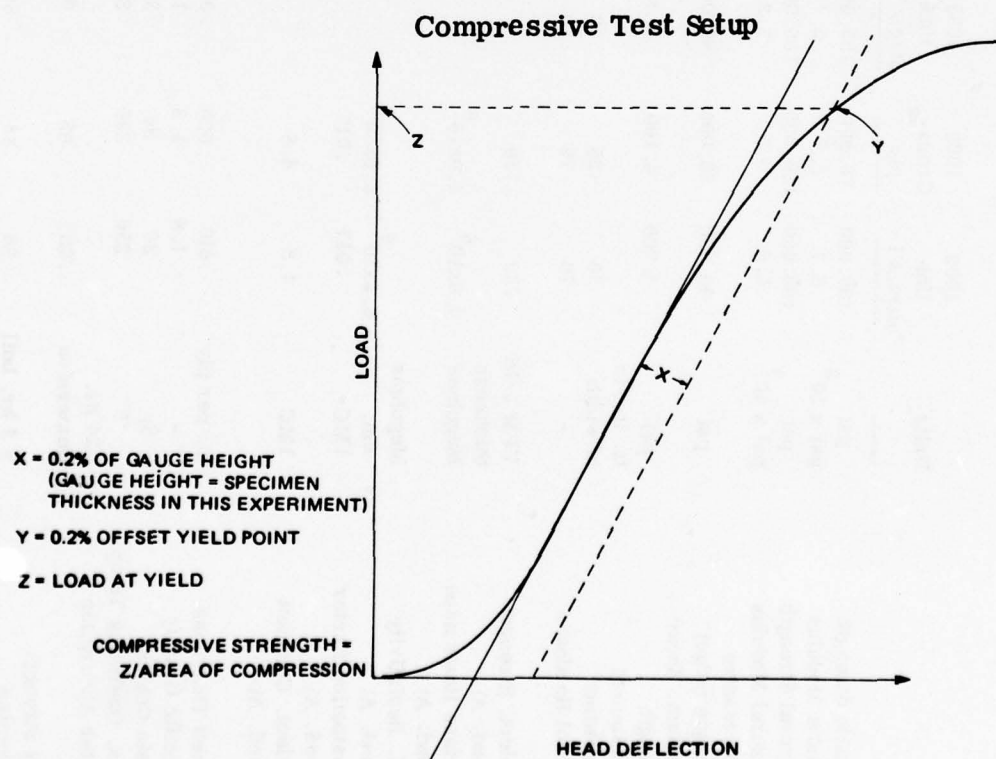
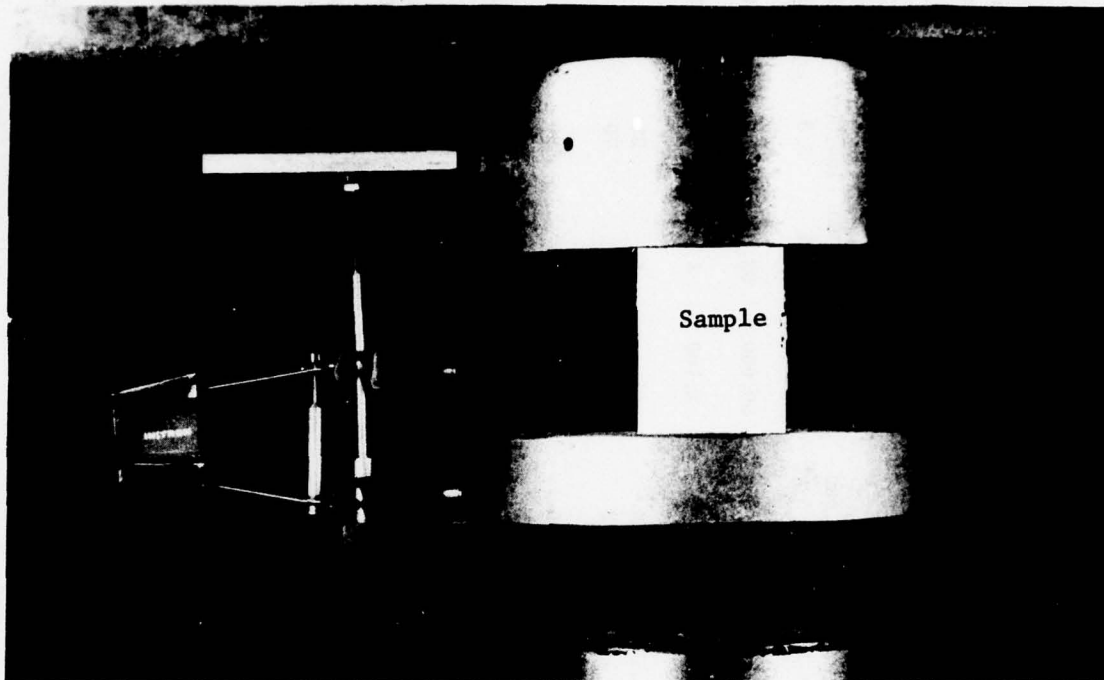


FIGURE 17. Typical Compressive Stress-Strain Curve

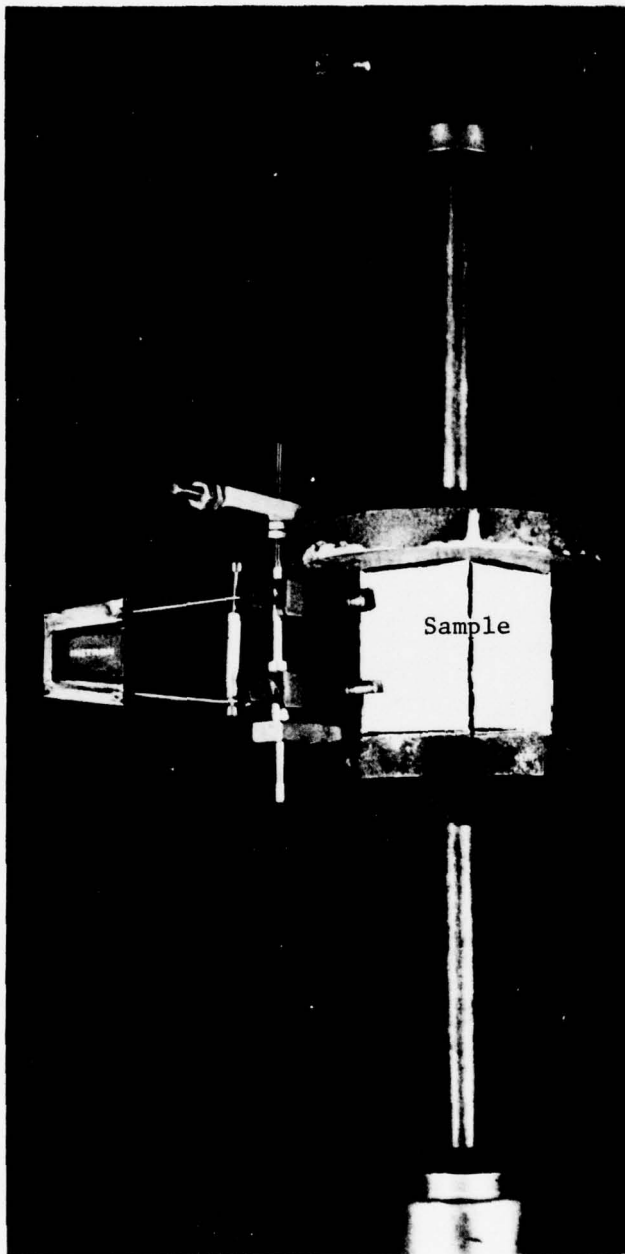


FIGURE 18. Tensile and Falling Ball Test Setups

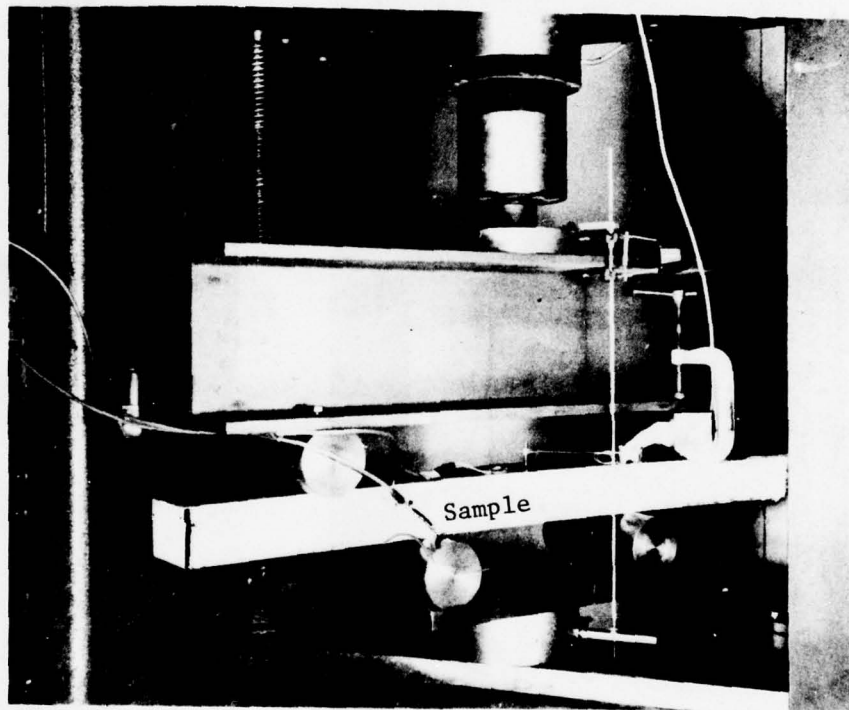


FIGURE 19. Bending Beam Test Setup

DATA SUMMARY

	1	2	3	4	5	6	7	8
Modulus (psi x 10 ⁻³)								
Compression, initial	32	157	217	126	179	135	108	136
After 7 days immersion*	37	156	216	124	183	133	105	133
Tensile	--	120	77	67	230	167	130	73
Yield (psi)								
Compression, initial	600	2150	2400	1750	1950	1700	1250	1529
After 7 days immersion*	625	1950	2450	1700	2150	1900	1250	1332
Tensile	--	315	250	230	460	335	318	355
Water absorption (% weight gain)								
24 hours immersion*	0.63	0.15	0.30	0.21	0.47	0.35	0.37	0.60
7 days immersion*	0.97	0.20	0.33	0.23	0.56	0.50	0.47	0.90
Density (lb/cu ft)	93.5	82.0	79.0	96.0	54.5	75.5	73.5	76.5
Specific gravity	1.50	1.31	1.27	1.54	0.87	1.21	1.18	1.22
Shear Strength (psi)	15	300	270	380	--	--	--	--
Shear Modulus (x 10 ⁻³)	26	66	59	66	109	70	67	63
Bulk Modulus (x 10 ⁻³)	57	105	59	130	103	125	102	97
Thermal expansion (in./in. /°F x 10 ⁷)	222	129	140	137	118	198	220	109

FIGURE 20

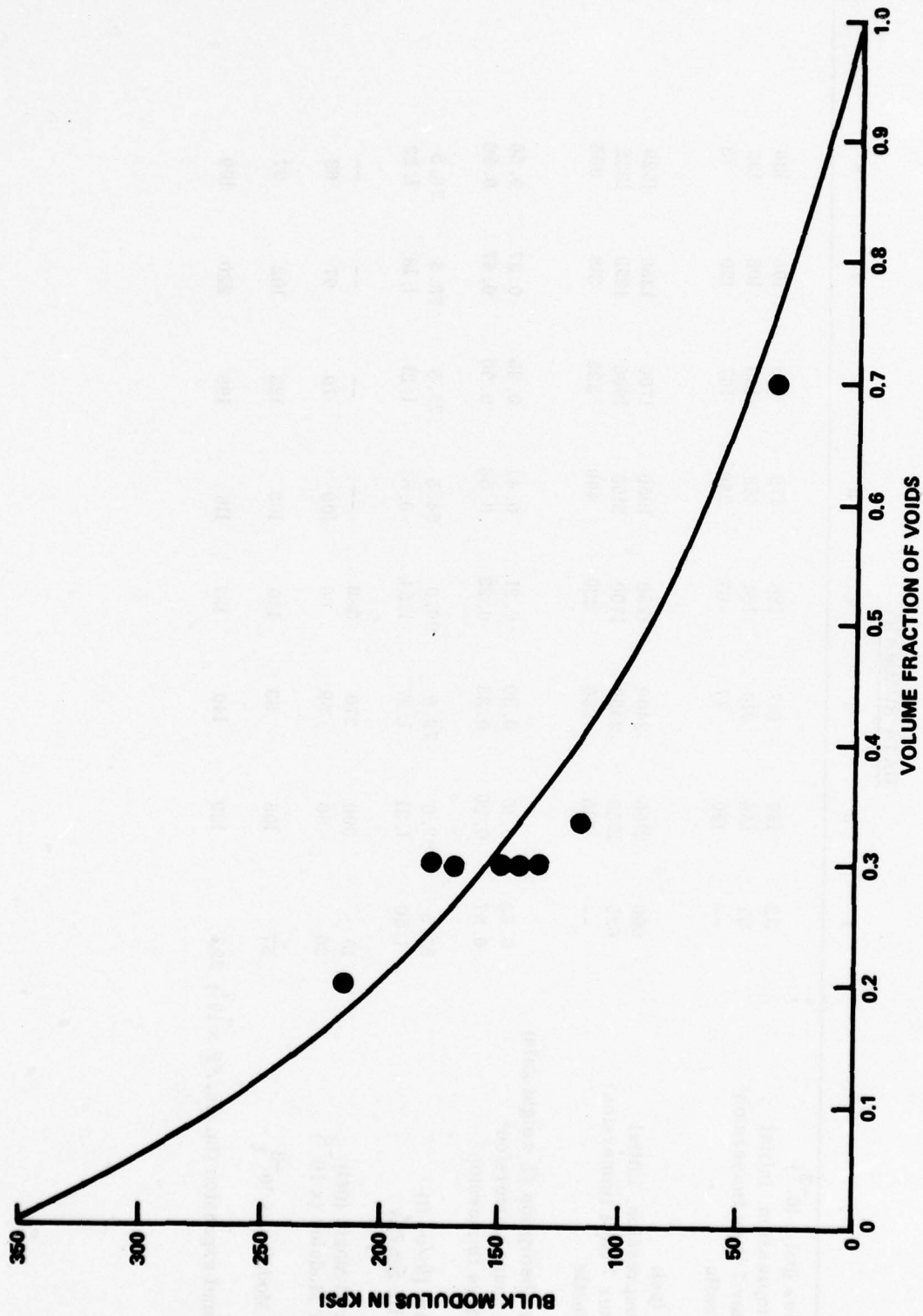


FIGURE 21. COMPOSITE SYNTACTIC FOAM - AIR VOIDS



NOTES FROM THE
DISCUSSION SESSION

Rapporteur (Rap) - The formal presentations made for a very logical introduction to this interesting, exciting field of acoustic attenuation materials systems.

Audience (Aud) - How much mismatch in acoustic impedance can be tolerated in a material so that the wave enters the system? For example, Dr. Madigosky presented some data which showed that his foams had sound velocities somewhere around 2500 meters/second. Sound speed in water is something like 1500 meters/second, yet apparently the wave enters the system. How much reflection do you get when you get that kind of difference in $\rho_0 c_0$?

Aud - The density of the foam is low so $\rho_0 c_0$ is very close to that of water so that there is not much reflection at normal incidence.

Aud - That is for normal incidence; what about for angular incidence? That is, how do we treat angular incidence?

Aud - The same numbers apply as in our earlier discussion. For example, if a surface had a matched impedance at normal incidence, then at 60 degrees off normal the impedance would be mismatched by a factor of 2. The corresponding pressure - reflection coefficient would be 1/3; that is an energy reflection coefficient of 1/9.

Aud - How about the linearity of the impedance? I noticed you were talking about 145, 150 dB or some values of that magnitude.

Aud - In airborne sound those are high levels and may be associated with non-linearities. However we are certainly not seeing non-linear behavior in most cases of interest in underwater sound.

Aud - How do you choose the impedance you want in service? What technique can you use for testing? Obviously it could also be ρc .

Aud - The one example that I spoke about was for airborne noise for a diffuse field. In such a case one purposely wants to choose a surface impedance about two times ρc for best absorption. In discussing off-normal incidence, the very simple picture I referred to was

for what is called a "point reacting surface", one where the local response to local pressure is the only thing that matters and what goes on elsewhere is not important. This means that the surface is non-wave bearing. If you have an interface between 2 fluids or a fluid and a continuous solid then your reaction is different.

Rap - Have we as theoreticians come far enough in the theory of attenuation or damping? Should we as an industry be doing more work theoretically or work supported by some empirical values of real, live situations? Can we design a system then that can do this complex function and if the manufacturing world and the materials world were up to analytical speed, could we indeed attenuate for infinite amount of power or not? Could we indeed make materials on a strength-to-weight basis that are orders of magnitude higher than where we are now? Do we need new theories to make an improvement?

Aud - As an example, consider the contribution of Sommerfeld to the anti-radar type of coatings for planes. It is not that it worked out all that terribly well but it involved interesting electromagnetic theory. That caused people to become interested in the possibility of radar screens. This is an example then where a well-known mathematical physicist starts a trend that is extraordinarily practical.

Aud - In the same sentence you use words like damping and attenuation. Those words are not interchangeable. Most of the sessions this morning were really concerned with mechanical damping processes and we heard very little about sonic attenuation materials in an absorbing process.

Rap - Is there a need to separate them? Can't we treat them together?

Aud - No, I think we have to treat them separately. The things you do in mechanical damping processes are neat mechanical tricks, but the absorption process is plain, simple absorption by physical processes. I don't mean to imply that it is that simple. Inherent within this whole thing are the processes which allow absorption to occur by viscous effects. Also the longitudinal wave entering from water separates into a p and s wave and each one propagates at a different velocity and thus interacts with different elements or particles in the medium. This could lead to scattering which could also be a very large absorbing factor in the overall mechanism. We really did not get into a

very serious discussion of that aspect of the energy absorbing process. I am merely implying that the term damping should be applied to a mechanically absorbing system whereas acoustic attenuation involves such effects as viscous losses, dispersion, and scattering.

Aud - Because the problem is that complex it probably involves all the mechanisms. One must be careful of the terminology used. Somehow or other we seem to slip very easily from damping to attenuation without knowing details of the material system and how it responds not only in the microsense but in the macrosense.

Rap - So your conclusion then is the need to isolate materials for various purposes.

As mentioned by one of the speakers, you not only have to have individual theory, and have to understand materials and process and safe design; you also have to treat a system. Perhaps what you are doing for ceiling tiles is different from what you do for wall tiles. Dr. Kerwin also mentioned that it was pretty straightforward to design for surface impedance for room-temperature sound absorbing. Is that complete unto itself, in that you know how loud a person can yell in a big room and you assume the room has a 50% relative humidity and a certain temperature? Are those parameters significant and should they somehow be addressed in the design of a panel for sound deadening? Or are these variables so small that it probably does not matter, and dear old trial-and-error and a little marketing are more important than determining the humidity in the room?

Aud - Humidity in medium-sized rooms is not a big problem. In very large amphitheatres it can be a problem for high frequencies. Performance peaks for absorbing materials for airborne sounds are pretty broad and essentially not amplitude dependent.

Aud - Suppose we had this room completely acoustically reflective and we put a sound source there. Assume then we made it completely acoustically absorbing at some reasonable distance from the sound source. What is the maximum sound pressure level difference? About 10 dB?

Aud - In the completely reverberant case, the free field case, what would be the difference in sound pressure level? I have heard people say 10 dB. We have made tests that sort of show that is true. Alternately, the air absorption is going to take over. Equilibrium will be reached someplace. Cases can be

found where one can get up to 20 but for realistic cases that is what we found. I would like to hear somebody tell me that is wrong.

Aud - What is the real difference between having damping or absorption or neither? What are we talking about in terms of quantitative numbers?

Aud - I think the answer is probably a lot more than 10 dB but I don't know what situation you had in mind.

Aud - Isn't the issue really that there isn't any perfectly reverberant surface and there isn't any perfectly absorbing surface? So, the answer to the question as stated, would probably be a tremendous number. In order to answer it intelligently one has to know how good your reverberant surface is and how good your absorbing surface is and perhaps with present technology the difference is 10 dB but with improved technology it might be 20, 50, whatever.

Rap - That is a good point. As the theory has it what we have now is nowhere near optimum. Therefore, the questions we have to address are: What is the industry doing or what should it be doing to lean toward optimum and lean away from inventiveness? Should we be looking at new theories, designing new materials? What is the way to move forward in this present industry for optima? What do we think as a community we have to do to talk in terms of 100 dB down instead of 5 dB?

Dr. Frisch gave a rundown on polymers and polyblends and presented some interesting ideas, especially in trying to design at the monomer/polymer stage a material that could provide properties that would be useful.

Aud - I wondered if you could not consider in the case of the crystalline polymers, the crystallites as being a high modulus filler and treat it in the manner that Dr. Nielsen did to explain the behavior of filled composition with "good" and "poor" adhesion of the polymer to the filler particles.

Aud - A few problems arise with crystalline polymers and that is that even if you apply relatively small deformations the Boltzmann and time-temperature superposition principles no longer apply. It is not easy to give a simple theory.

On the other hand, if you are saying as a first approximation one can account for small volume

fractions of crystallites, yes, perhaps as much as roughly 20%.

Aud - There are all kinds of ways dispersion can occur in polymer systems. If you want to use viscoelasticity as a mechanism for getting the energy out of the wave, that is one of the ways of doing it. What I am concerned about is, of course, is just how much you can get out for a given thickness of material. How big is the attenuation coefficient?

Aud - You could form an IPN from a very elastic material and a glassy material with T_g 's very much separated. This would be a good IPN. A morphology with domain size on the order of 80 Å or less, I would expect would give broadening over an extended range of temperatures and frequency. One might work with a controlled morphology to produce this kind of scattering phenomenon. Why not? By working with different compositions of the materials you can control the morphology.

Aud - We made pseudo IPN's out of polystyrene and crosslinked ethylene/propylene copolymer rubbers; and polystyrene and crosslinked ethylene/vinyl acetate copolymer rubbers. Perhaps these have the kind of scattering properties one wants.

Aud - Has any work been done to define on the basis of T_g separation what the domain size must be for optimum T_g or loss factor broadening? It seems to me there has got to be an optimum. Above an optimal size they will behave like two independent materials. If you go down to the point where you start totally losing the domain morphology, what is going to happen? There has to be a morphology or gradation of domain morphology.

Aud - As far as domain size is concerned it may range down to 100 Å or less as you get down to very fine particle size. In the case of IPN's much smaller size particles result as compared to pseudo-IPN's or simple polyblends.

Rap - From a chemical viewpoint, is a polymer blend sensitive to batch size, quantity, mass? My experience has been that as you go from laboratory size to pilot plant size to production size, new things are uncovered. Somehow changes result; the product gets implemented because so much effort is put into it. However, it never really meets its forecasted goal. I was wondering if there were some better understanding of theory or better understanding of design, or better

understanding of service conditions that would make it meet the goal.

Dr. Nielsen of Monsanto gave us a good review of the mixture of mechanical properties and damping in syntactic foams. He gave us a fine rundown on fillers versus intrinsic shear modulus along with temperatures and mentioned a few dichotomies. Some of us think they were dichotomies, others think they were well understood but never publicized.

Aud - I am intrigued with the idea of platelets. If, for example, you take mica or graphite or things of that sort which are micro- or multi-platelet materials as compared to a more homogeneous material in platelet or flake shapes; is there a marked difference in behavior performance from an acoustical damping standpoint?

Rap - How does one determine what filler is needed?

Aud - Platelet fillers are unique in their effect in broadening the transition zone and improving the damping over a range of temperature. The effect also seems to be independent of frequency.

Now we have always attributed this enhancement of the damping to the slippage of graphite platelets, like the cards in a deck of cards. YOU cannot breakdown one of those graphite "agglomerates" to its ultimate single platelets in any kind of processing that I know of. Incidentally, although both have the platelet structure graphite is better than mica. We also tried several other things as model materials, but they did not perform nearly as well. We tried flaked glass, glass fibers, spherical particles, particulates of different densities, powdered metals, and organic polymer latex particles. None of these performed nearly as well as the platelet fillers. So we went back to the platelet fillers (graphite), which gave a big improvement over a range of frequencies, and studied these as a function of type of graphite, of loading etc. Small amounts of carbon black, with the graphite, provided some further improvement in damping performance.

Rap - Was there a good referee test to determine when it performs well or when it does not, or was there some question about that?

Aud - We measured free bar, and constrained damping over the specified range of temperature and pressure and compared the results with controls. Measurements were made over the range of 100-1000 Hertz.

Aud - I wonder if in the process for preparing these materials, viscosity perhaps is such that these particles of flakes will randomly orient themselves or do they tend to clump and layer and thereby show an anisotropy?

Aud - Not that we could tell. We made some scanning electron microscope photos and the particles do not seem to be oriented. The graphite particles are pretty small to begin with. In looking at our prepared composites with an electron scanning microscope the graphite platelets did not seem to be characteristically oriented.

There was another thing, however. There is a very big difference, for example, between the poly(vinyl chloride) and vinyl chloride/vinyl acetate copolymer plasticized to put the Tg's in exactly the same range. The vinyl chloride/vinyl acetate copolymer performs better than the comparably plasticized poly(vinyl chloride) by a very considerable margin. We think that this may have something to do with the fact that the straight poly(vinyl chloride) has some crystallinity, and this has an effect of decreasing the maximum in the damping. Although this may provide some broadening effect this is offset by the big loss in maximum damping. We did not need the broadening; we were getting that out of the filler anyway.

Aud - Transcrystalline polymer behavior depends on how you mold the sheet or plate particularly against a very high surface energy nucleating medium, for example, certain metals. You get a proximal layer in which your spherulites instead of being randomly oriented in the material are aligned. Of course, this is going to produce a very interesting effect. It is going to give you anywhere from 10 microns to 80 microns of the transcrystalline region. This can be especially useful if you want to work with polycrystalline material.

Aud - Platelet materials have very large distances in the "C" direction as in graphite for instance, which leads to a very large compressibility. When you try to get energy out by viscoelasticity, you are dependent upon the bulk modulus. The bulk modulus in going from the glassy state to the rubbery state changes only by a factor of three-to-four, whereas Young's modulus changes by a factor of 1,000. So what you need to do to get energy absorption is to provide a way of getting translation of compressibility. You do not normally get that unless you put something into the material to introduce a very large free volume which provides the

necessary dynamic compressibility. Then you have a system that really can absorb the energy.

Aud - It seems to me that the name of the game is designing the system to get the energy into the energy absorbing material. The distinction between attenuation and vibration damping disappears when you think of it in this way. One comment was made about how energy is absorbed in materials. Some of the rest of us were talking about how a structure is designed to properly incorporate the damping material, so that the energy is efficiently transferred to the material. The latter is a different subject and the key to understanding and using the whole concept. You may have a very efficient damping material, but if you do not design your system to work it efficiently you have not done anything. In our experience most unsuccessful attempts to dampen or attenuate have fallen down in this area.

Rap - I would just like to add one more thing to that. You have to pass this information in an understandable, readily readable form or the designers of the equipment are going to miss it.

Aud - But, I don't think the absorption of the energy is any different in either case since the energy ultimately is transferred to the absorbent material. Both are going to end up in a wave being propagated through the material, and that way the material resists propagation and that is damping. The difference between acoustic damping and structural damping is the way the energy is transferred into the material.

Aud - Is it true that, generally, low frequency is much more difficult to optimize for, than high frequency? Maybe we could concentrate on different absorbing mechanisms and/or materials for each range of frequency, then combine them to produce a useful combination. In other words, we again have an example where the useful engineering materials are usually very heterogeneous rather than homogeneous.

Aud - I have a question concerning the role of adhesion with flaked graphite. Did you do any work specifically with the adhesion of these graphite particles? Do you have any indication that a bonded graphite particle was any different than one that was treated to minimize adhesion.

Aud - Dr. Nielsen did and in fact I think the most important thing that he showed this morning was that you could use these two different types of silane

materials either to promote or discourage adhesion. I do not think that he applied that to graphite. The other point he mentioned that I thought was most important was the differential thermal coefficient of expansion between the filler and the matrix material. The effects of thermal coefficient of expansion represent an area we would like to explore some more ourselves.

Aud - One other point I would like to clarify. This gentleman over here thinks everything is eventually damped, in all cases. Whether you try to attenuate or whether you try to damp, it all results in damping. Can't you also accomplish attenuation by complete dispersion? You do not actually absorb the energy, you reflect it away at a different angle. The importance of each mechanism should vary from case to case.

Rap - Why don't we talk a little bit about shaping factors then. Are they real, do they make sense? Do you look at a mixture and do some shaping, do some absorbing, do some reflecting? First of all do we like to make cone-like shapes in an anechoic chamber?

Aud - I want to look at a cone or parabola of rotation that had the bottom out where the focus is beyond the bottom. Getting the energy out of the water it can then enter some medium where you can absorb it. I do not know if this has been considered or not. The analogy that comes to mind is with the compound parabolic collector used in solar collectors. Now once you have got it through and presumably concentrated, you should be able to absorb it more effectively. Has this ever been explored at all?

Aud - This is akin to surface waves in whispering galleries, e.g. in the Mormon Tabernacle in Salt Lake City.

Rap - In my company we have an anechoic chamber that we put a complete airplane in to minimize our side lobes, and that is why that chamber was designed that way. I have need to go in there every now and then and the sound indeed is low, but it was designed for RF. Does this shape work on whatever the wave form is, in whatever medium it is in? Has the industry looked at it from that viewpoint? Do striations and shear layers help, hurt, or just cause complications? As pointed out in one of the other talks, this is a weight sensitive, cost sensitive environment we are in. We have got to have some better answers than we have now.

Aud - There was something that I did not mention this morning when I showed the slide on wedges. Professor Meyer, under whose guidance that particular work was done, was also interested in making a multipurpose chamber. He had the glass-fiber wedges soaked in a solution containing graphite, making them partially conducting. Thus the chamber was absorptive for electromagnetic waves as well as acoustic waves.

Regarding shape and absorption, one can separate the problem of absorbing by a gradual impedance transition into two functions. One of which is to "trick" the wave, subvert it gradually, get it into some layer where you can then absorb it. That might be the analog of your double parabolic device to focus energy so you can absorb it. On the other hand, if it is just as easy, you may be able to absorb it while you are "tricking" it. The final choice of absorbing configuration will depend on questions like, "Can you stand the dimensions? Do you like the surface? What's the cost?" etc.

Rap - I could not help but think when I was listening to Dr. Nielsen as he showed the variations expected and unexpected of the atomized aluminum powder-filled thermoplasts that one of the conclusions, after the fact, was that mismatch of linear coefficient of thermal expansion could have been the cause or the difference. My question is -- could you not consider that before you begin? Could you not consider thermal diffusivities? Could you not consider other parameters prior to running the experiment or do you run a series of experiments and then make conclusions?

Aud - I think the answer is yes? I would do both. It seems to me that the efficient and well-proven way in these complex fields is to get into the right ballpark with some experimental tests. Then it seems best to determine properties of the individual constituents and try to weigh and combine them in the right proportion to explain the results of the tests and to help predict the future directions.

It would be nice to skip the first pragmatic step, but it does not seem to happen very often.

Rap - We heard a talk on materials and processing. Are there any questions from the floor as it relates to syntactic materials and processes?

Aud - Is there a problem of NDT in this area? Do people know what has to be specified and controlled to guarantee that they are getting reproducible material?

Rap - From my vantage point, I think the rigorous answer is no. However, subjectivity still remains and in a rigorous classic sense you can not just use one non-destructive test and be certain. You are more certain if you integrate control of what material you are starting with, how you are preparing the material in its final form, and then apply non-destructive testing. It is a more complete quality control. The final answer is operational service free of mishaps which tells you several years later that indeed your quality procedure was adequate.

Dr. Tweedie gave us a good insight into many of the things that are happening in spacecraft and I believe that everything he talked about today is universal to all kinds of crafts. Therefore, it should have a great interest across the board. But the thing that he mentioned that was particularly significant, because I come from the aerospace industry, is weight. Whatever job you are asked to do, whatever field of endeavor you are in, the person that does that job well is the person that does it at minimum weight. Because, weight relates to cost -- relates to time -- relates to numerous things.

Aud - Some of the examples given in Dr. Tweedie's talk illustrated the fact that the sound absorbing or damping material considerably increased the stiffness of the part, nevertheless no effort was made to take advantage of this for structural purposes. It seems to me for instance, if I had been designing those printed circuits I might have made them on a thinner substrate since after all, part of the stiffness was contributed by sound absorbing material. Then the next time I would have gone a step further and see if I could make the substrate sound absorbing at the same time and not require the extra operation. Why didn't you do that?

Aud - I would very much like to do that, but the real situation however, is that most of the time people who design structures, especially electronic structures, do not pay any attention to the vibration problem until they test it and find out it does not work and have to come and ask someone to fix it. In almost every example there was that kind of after-the-fact situation.

Rap - Regarding at the structure around the engines, we are constantly upping the thrust and noise levels. I believe on the B1 we are up to 167 dB, considerably higher than the threshold of pain. It is creating all kinds of structural problems and there are indeed various design techniques that are being studied, but

they only are studied because, as mentioned before, they become an operational problem. They are not addressed early enough and I guess that is what some of us ought to be doing now, looking at that without controlling the structures engineer or the aerodynamicist but learning to work with them.

The last paper of the day was by Dr. Madigosky and I think the bottom line of his discussion was that the law of mixtures worked on mixtures. Is the law of mixtures all you need to know, or is it just a start?

Aud - The statement was made that the law of mixtures works on mixtures. That seemed to be true as far as the properties of syntactic foam itself. At the end of the presentation, a curve was shown of the mechanical properties of a syntactic foam filled with randomly shaped and sized pieces of viscoelastic material. A curve was displayed which apparently was calculated and I was not able to figure out whether it showed any comparison with actual real life experimental data.

Aud - There were eight samples which were constructed and tested. They had various loadings of the particles and inclusions, but they were all on the order of 30 percent or so. If you look at the test results, most of the moduli for those samples were on the order of 100 to 150 thousand psi, and that was about what the theory in the last graph predicted.

Aud - In other words the theory is correct at one point, which any good one parameter theory should be able to do?

Aud - Yes, I did not really have specific data on the samples to give me enough parameters to go down sample by sample.

Aud - The point I started to make is, really, as far as the last argument, is the data really sufficient to make a conclusion? The simple formula fits at one point at a loading of 30%.

Aud - My point was really that, here is a theory or something that gives some information, and it looks as if it works. It gives at least the right order of magnitude of the effects and it might be useful. That was I think the point.

Aud - Isn't there more than one law of mixtures?

Rap - There are an infinite set of laws of mixtures. I got involved in matters where laws of mixtures were

pretty important. It goes back to the days of filament wound fiberglass. Today it is at a stage of advanced composites, advanced primary structure, and the root of the problem with most of these orthotropic and now anisotropic high performing materials is that loss of the particular structure in which it appears is loss of the system. By not starting with laws of mixtures, design curves, you can not start out at a complex computer program using finite element analysis or what have you, unless you can bound the problem. I think what Dr. Madigosky did is good; that is, create the first experiment and then optimize by the use of any analytical tool to move forward in the understanding of this very complex material, which I think we all agree is complex.

Aud - Did you consider treating these things with just random shaped holes with no material in it?

Aud - No, I threw in a guess for the elastic properties of the inclusions, but the difference in moduli between the polymer and the inclusions is large enough so that the exact value of that guess is very unimportant.

Aud - The other thing that bothered me about the theory for that last material you discussed is that the comparison was based only on the real modulus. With reference to the subject of today's workshop, we have to know the complex modulus. We have to know the loss modulus as well as the real part. Of course, there are theories, by Hashin and others, that attempt to do that. Did you try to apply the same approach to calculate not only the real modulus but also the imaginary part, and do you have any idea whether that fits or not?

Aud - I have not done it for this system.

Rap - Are there any other questions before I thank you all for some very good staying power, and some very frank discussions. I think from a workshop point of view, it was quite a workshop.

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